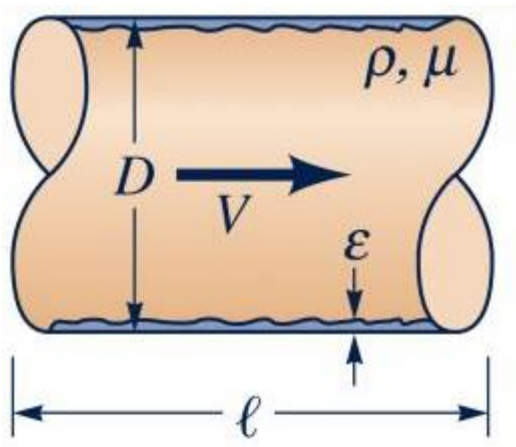
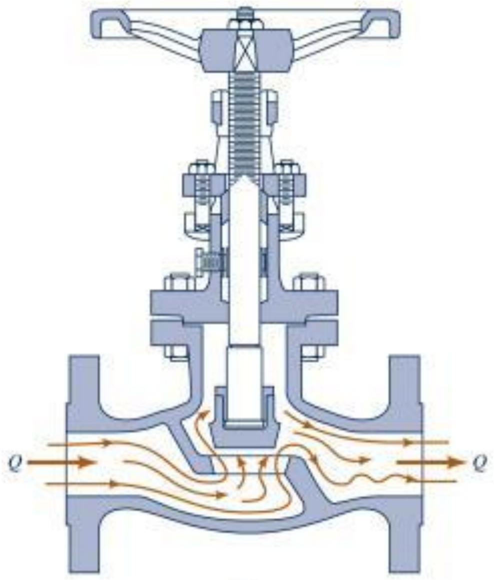
Lecture 6

Steady Flow in Pipes (3)





Contents

Objectives

- Lean how to determine the friction factors for commercial pipes
- Study friction factors for non-circular pipes
- Study empirical formulas
- Determine the local loss due to the shape change of pipes.

6.1 Pipe friction factors

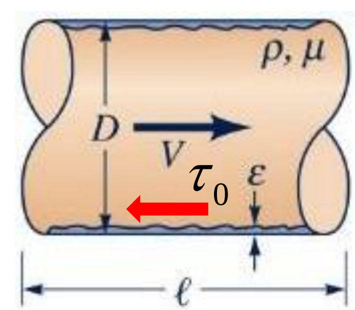
- Since there is no exact solution for the pipe friction factors, determination of friction depends extensively on the experimental works.
- Dimensional analysis

- Wall shear stress depends on the mean velocity, pipe diameter, mean roughness height, fluid density and viscosity.

$$f(t_0, V, d, e, r, m) = 0$$

- $n = 6$;
- $k = 3$ r : Mass, V: Time, d: Length

$\rightarrow \pi$ (non-dimensional term) = 3



6.1 Pipe friction factors

- Then $\pi_1 \Rightarrow \phi_1(\rho, V, d, \tau_0)$

$$\pi_2 \Rightarrow \phi_2(\rho, V, d, \mu)$$

$$\pi_3 \Rightarrow \phi_3(\rho, V, d, e)$$

$$M^0 L^0 t^0 = \phi_1(\rho, V, d, \tau_0) = \left[ML^{-3}\right]^a \left[Lt^{-1}\right]^b \left[L\right]^c \left[ML^{-1}t^{-2}\right]^{-1}$$

$$a - 1 = 0, \quad -3a + b + c + 1 = 0, \quad -b + 2 = 0$$

$$a = 1, \ b = 2, \ c = 0$$

$$\pi_1 = \phi_1\left(\frac{\rho V^2}{\tau_0}\right) = \phi_1\left(\frac{\tau_0}{\rho V^2}\right)$$

- In the similar way,

$$p_2 = f_2\left(\frac{Vdr}{m}\right), \quad f_3 = f_3\left(\frac{e}{d}\right)$$

6.1 Pipe friction factors

- Therefore

$$\frac{\tau_0}{\rho V^2} = \phi\left(\frac{Vd\rho}{\mu}, \frac{e}{d}\right) = \phi\left(\mathrm{Re}, \frac{e}{d}\right)$$

$$\tau_0 = \rho V^2 \phi'\left(\mathrm{Re}, \frac{e}{d}\right) \qquad \left(remember,\ \tau_0 = \frac{f\rho V^2}{8}\right)$$

$$\text{Finally} \quad : \quad \boxed{f = \phi''\left(\mathrm{Re}, \frac{e}{d}\right)}$$

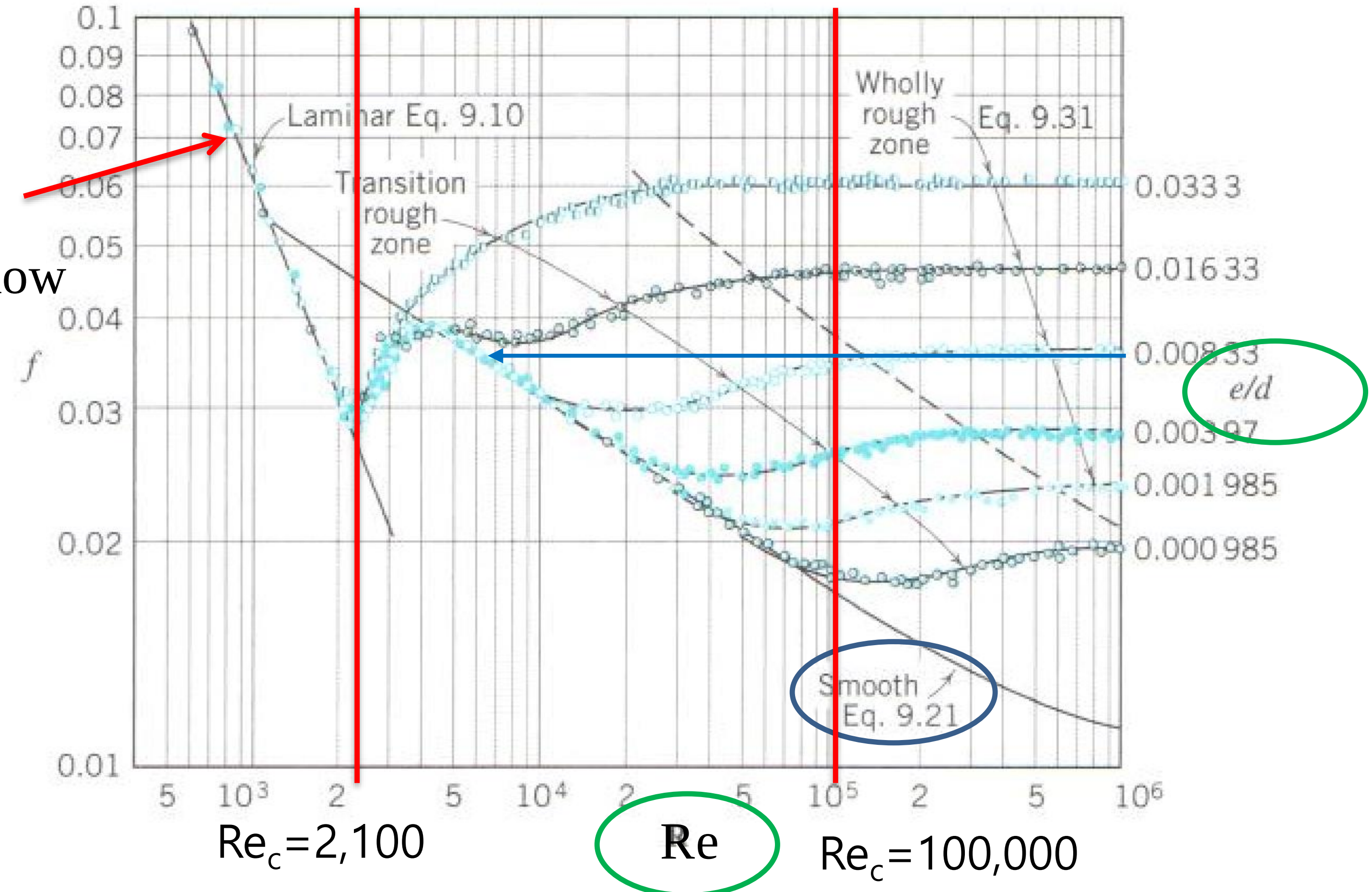
If Reynolds number, roughness pattern and relative roughness are the same (If dynamically and geometrically two systems are same), then their friction factors are the same.

P.R.H. Blasius (1883~1970; German)

- **Blasius (1913) - Stanton (1914)** suggested the diagram based on Nikuradse data

$$f = \frac{64}{Re}$$

Laminar flow



Re_c=2,100

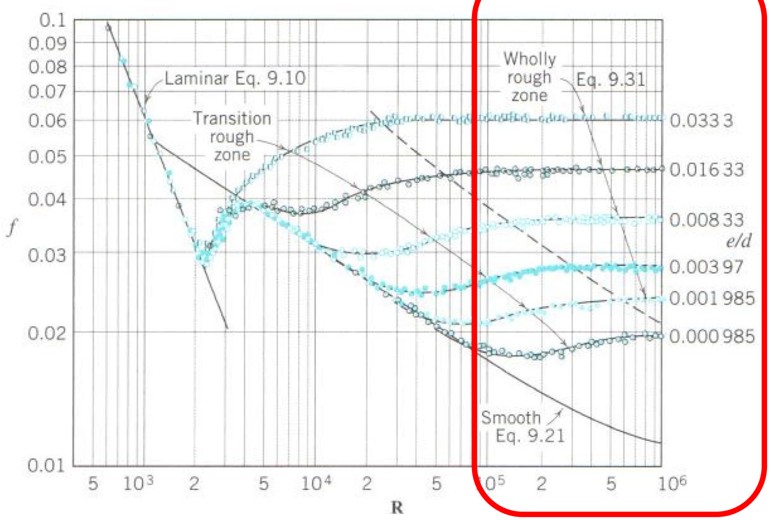
Re_c=100,000

1) For laminar flow ($\text{Re} \le 2{,}100$), $f = \dfrac{64}{\text{Re}}$

2) In turbulent flow, a curve of *f* versus *Re* exists for every relative roughness, *e/d*.

- For rough pipes, the roughness is more important than *Re* in determining *f*.
- At high Re (wholly rough zone), *f* of rough pipes become constant dependent wholly on the roughness of the pipe.

$$\frac{1}{\sqrt{f}} = 2.0 \log \frac{d}{e} + 1.14 \qquad (6.1)$$

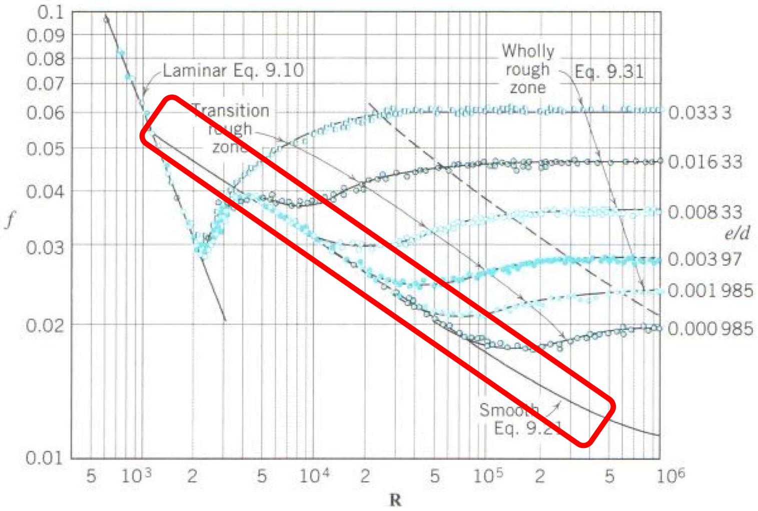


- In turbulent flow of smooth pipes, *f* is given by

$$\frac{1}{\sqrt{f}} = 2.0\log\left(\mathrm{Re}\sqrt{f}\right) - 0.8 \qquad (6.2)$$

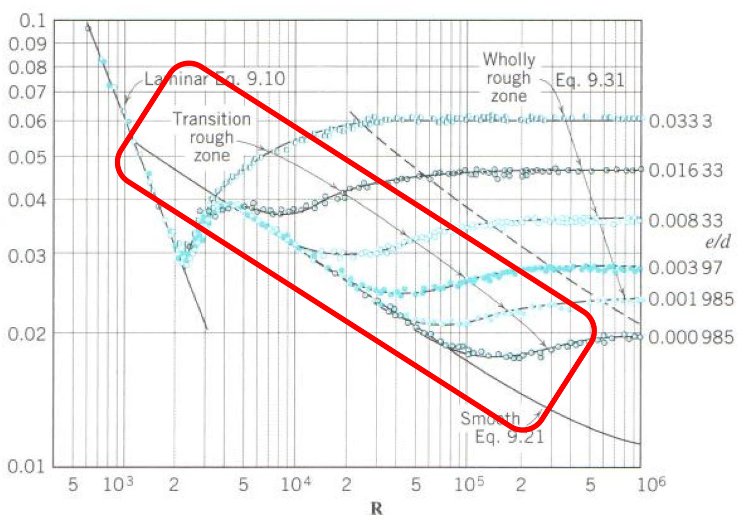
$$f = \frac{0.316}{\mathrm{Re}^{0.25}} \text{ by Blasius} \qquad (6.3)$$

- For smooth pipes, the roughness is submerged in the laminar sublayer, and have no effect on *f*.



- In turbulent flow of transition zone, the series of curves for the rough pipes diverges from the smooth pipe curve as *Re* increases.
- Pipes that are smooth at low values of Re become rough at high values of *Re*. (The thickness of the laminar sublayer decreases as *Re* increases)

Re	Smooth	*e/d*= 0.0019	*e/d*= 0.0083	*e/d*= 0.0163	*e/d*= 0.0333
10,000	0.031	0.031	0.032	0.037	0.055
20,000	0.026	0.026	0.030	0.042	0.058
50,000	0.021	0.022	0.033	0.046	0.060
100,000	0.017	0.023	0.036	0.047	0.060



IP 9.8; pp. 346-347

- Water at 100°F flows in a 3 inch pipe at a Reynolds number of 80,000. If the pipe is lined with uniform sand grains 0.006 inches in diameter,
  1) How much head loss is to be expected in 1,000 ft of the pipe?
  2) How much head loss would be expected if the pipe were smooth?

1) Transition

$$\frac{e}{d} = \frac{0.006}{3} = 0.002 \text{ and } \mathrm{Re} = 80{,}000$$

→ Transition between smooth and wholly rough condition

$$f \cong 0.021 \quad \text{(Use Blasius-Stanton diagram)}$$

$$V = \frac{\mathrm{Re} \times \nu}{d} = \frac{80{,}000 \times 0.739 \times 10^{-5}}{3/12} = 2.36\, ft/\sec$$

$$h_L = f\frac{l}{d}\frac{V^2}{2g_n} \approx 0.021\frac{1000}{3/12}\frac{2.36^2}{2 \times 32.2} = 7.3\, ft$$

2) Smooth pipe

If flow is in <u>a smooth pipe</u>, then we can apply Blasius power relationship (Eq. 5.14)

$$f = \frac{0.316}{\mathrm{Re}^{0.25}} = 0.0188$$

The head loss in the smooth pipe

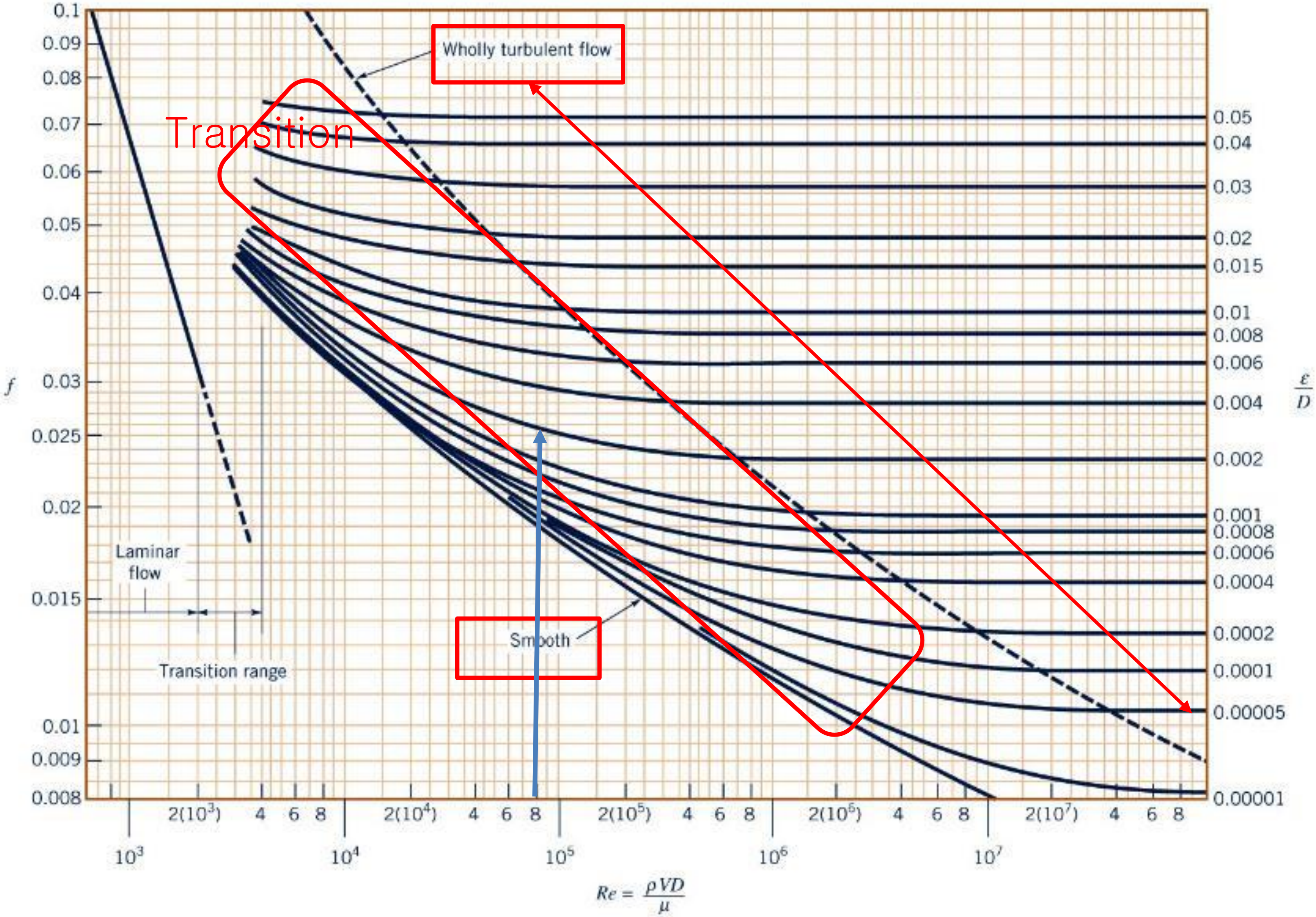
$$h_L = f \frac{l}{d} \frac{V^2}{2g_n} = 0.0188 \frac{1000}{3/12} \frac{2.36^2}{2 \times 32.2} = 6.5\,ft$$

Moody Diagram

L.F. Moody (1880~1953; US)

- Colebrook showed that Nikuradse's results were not representative of commercial pipes.
- Roughness patterns and variations in the roughness height in commercial pipes resulted in friction factors which are considerably different that Nikuradse's results in the transition zone between smooth and wholly rough turbulent flow.
- **Moody (1944)** presented the Colebrook equation in graphical form using Blasius-Stanton format.

→ Moody diagram along with *e*-values for commercial pipes



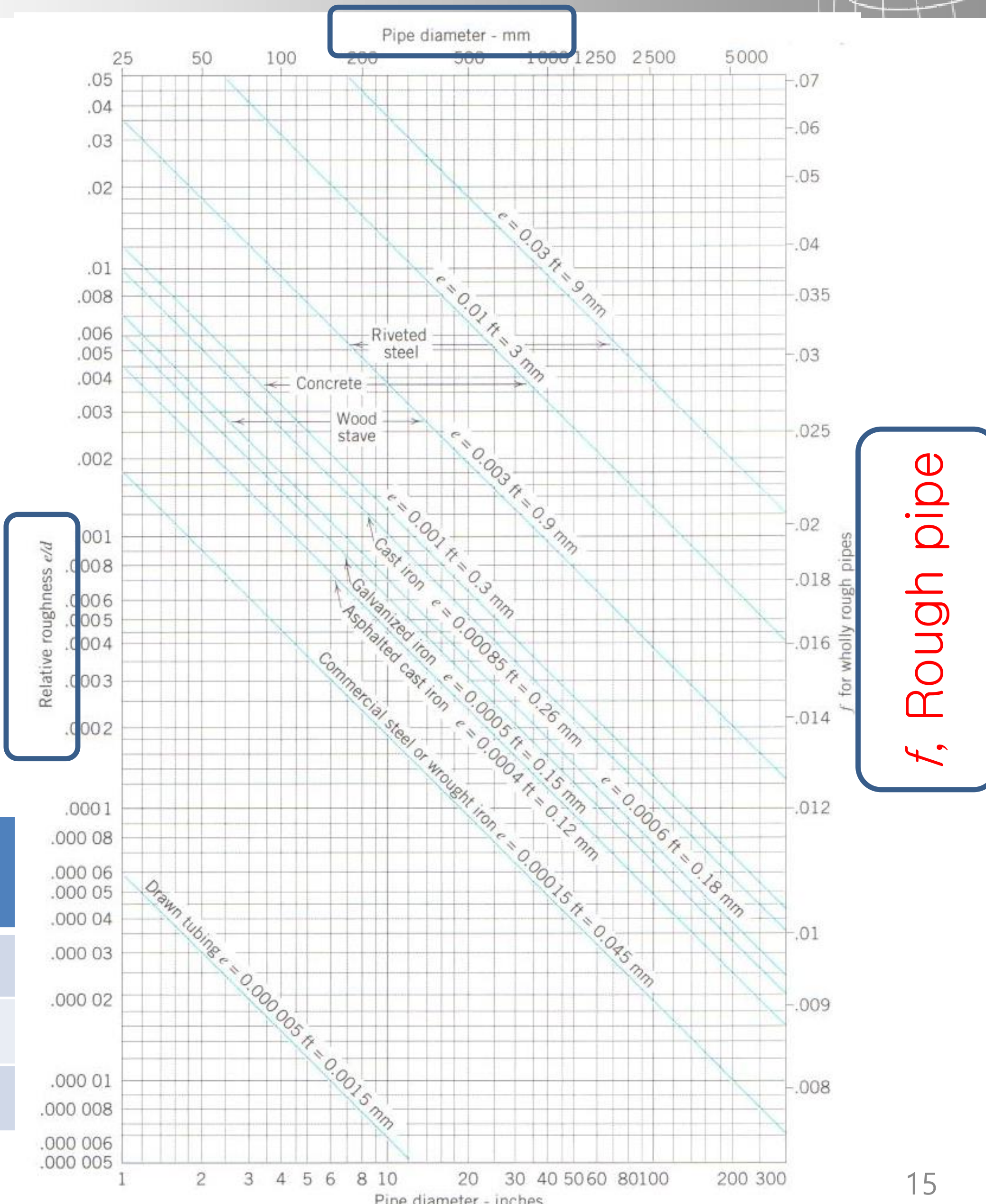
Wholly turbulent flow
Transition
Laminar flow
Transition range
Smooth
f
ε/D
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.025
0.02
0.015
0.01
0.009
0.008
0.05
0.04
0.03
0.02
0.015
0.01
0.008
0.006
0.004
0.002
0.001
0.0008
0.0006
0.0004
0.0002
0.0001
0.00005
0.00001
10^3
$2(10^3)$
10^4
$2(10^4)$
10^5
$2(10^5)$
10^6
$2(10^6)$
10^7
$2(10^7)$
4 6 8
$Re = \frac{\rho V D}{\mu}$

Moody Diagram

- The relative roughness should be determined by Fig. 9.11 (p. 349).

- Cast iron: 주철(주물)
- Galvanized iron: 도금강
- Wrought iron: 연철(단조)
- Drawn tubing: 압연튜브

d, mm	Steel	GI	CI	Concrete
200	0.014	0.018	0.020	0.029
500	0.012	0.014	0.016	0.023
1,000	0.015	0.012	0.014	0.020



Moody Diagram

- DARCY (pp. 713-716)

- A computer program which calculate a Darcy-Weisbach friction factor for a given Reynolds number and relative roughness.

- The roughness of commercial pipe materials varies widely with the manufacturer, with years in service, and with liquid conveyed.

- Corrosion of pipe wall material and deposition of scale, slime can drastically increase the roughness of the pipe and the friction factor.

Unnumbered 8 p429a
Photograph courtesy of CorrView

IP 9.9; p. 350

- Water at 100°F flows in a 3 inch pipe at a Reynolds number of 80,000. This is a commercial pipe with an equivalent sand grain roughness of 0.006 in. What head loss is to be expected in 1,000 ft of this pipe?

$$\frac{e}{d} = \frac{0.006}{3} = 0.002 \text{ and } \text{Re}{=}80,000$$

$$f \cong 0.0255 \quad \text{(Use Moody diagram)}$$

$$h_L = f\frac{l}{d}\frac{V^2}{2g_n} \cong 0.0255\frac{1000}{3/12}\frac{2.36^2}{2\times 32.2} = 8.8\,ft$$

→ This is 30% higher than previous result (7.3 ft) for the pipe lined with real sand grains.
However, under smooth pipe and wholly rough conditions, both pipes would have the same head loss.

6.2 Pipe friction in noncircular pipes

- Friction factor and head loss in rectangular ducts and other conduits of noncircular form.
- Use hydraulic radius (동수반경)

$$R_h = \frac{A}{P} \qquad (P \text{ is wetted perimeter, } A \text{ is area})$$

- First calculate the hydraulic radius and determine the equivalent diameter of the circular pipe.

$$d = 4R_h \qquad \left(R_h = \frac{\rho R^2}{2\rho R} = \frac{R}{2} = \frac{d}{4} \right)$$

- Use this diameter for Moody diagram.

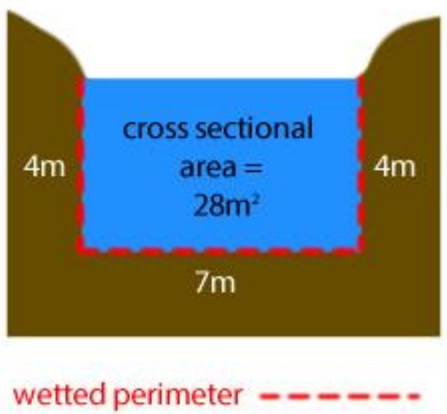
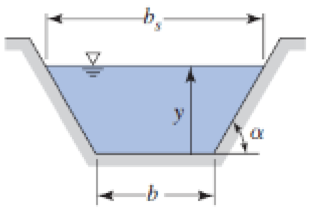
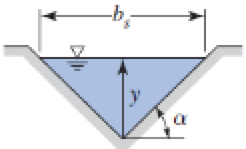
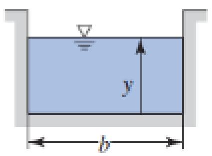
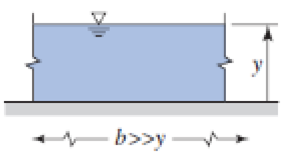
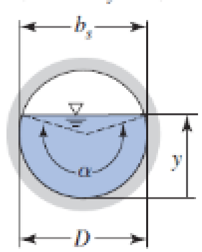


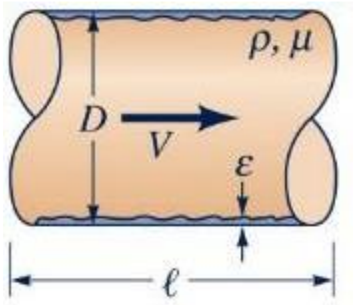
Table 11.1

Geometric Properties of Common Open-Channel Shapes

Shape	Section	Flow Area, A	Wetted Perimeter, P	Hydraulic Radius, R_h
Trapezoidal		$y(b + y\cot\alpha)$	$b + \frac{2y}{\sin\alpha}$	$\frac{y(b + y\cot\alpha)}{b + \frac{2y}{\sin\alpha}}$
Triangular		$y^2\cot\alpha$	$\frac{2y}{\sin\alpha}$	$\frac{y\cos\alpha}{2}$
Rectangular		by	$b + 2y$	$\frac{by}{b + 2y}$
Wide Flat		by	b	y
Circular		$(\alpha - \sin\alpha)\frac{D^2}{8}$	$\frac{\alpha D}{2}$	$\frac{D}{4}\left(1 - \frac{\sin\alpha}{\alpha}\right)$

- In turbulent flow, hydraulic radius concept seems work but in laminar flow not applicable.
- Darcy-Weisbach eq.

$$h_L = f\frac{l}{d}\frac{V^2}{2g_n} = f\frac{l}{4R_h}\frac{V^2}{2g_n}$$



- Hydraulic radius is reciprocal to the wetted perimeter *P*.

Thus, head loss is proportional to *P*, which is index of the extent of the boundary surface in contact with the flowing fluid.

In turbulent flow, pipe friction phenomena are confined to thin region adjacent to the boundary surface.

However, in laminar flow, friction phenomena results from the action of viscosity throughout the whole body of flow.→ large errors

IP 9.11; pp. 352-353

- Calculate the loss of head and the pressure drop when air at an absolute pressure of 101.kPa and 15°C flows through 600 m of 450 mm by 300 mm smooth rectangular duct with a mean velocity of 3 m/s.

$$R_h = \frac{A}{P} = \frac{0.45m \times 0.30m}{2 \times 0.45m + 2 \times 0.30} = 0.090$$

$$\mathrm{Re} = \frac{Vd\rho}{\mu} = \frac{V\left(4R_h\right)\rho}{\mu} = \frac{3m/s \times \left(4 \times 0.090m\right) \times 1.225kg/m^3}{1.789 \times 10^{-5}} = 73{,}950$$

$$f \cong 0.019 \quad \text{(From the Moody diagram for smooth pipe)}$$

$$\mathrm{h}_L = f\frac{l}{d}\frac{V^2}{2g_n} = f\frac{l}{4R_h}\frac{V^2}{2g_n} = 14.5m$$

$$\Delta p = \gamma h_L = \rho g h_L = 174kPa$$

6.3 Empirical Formulas

The Darcy-Weisbach equation provides a rational basis for the analysis and computation of head loss.

$$h_L = f\frac{l}{d}\frac{V^2}{2g_n} \rightarrow V = \sqrt{\frac{2g}{f}}d^{1/2}S_f^{\ 1/2} \qquad S_f = \frac{h_L}{l}$$

1) **Hazen-Williams** (1933): turbulent flow in a smooth pipe

$$V = 0.849C_{hw}R_h^{0.63}S_f^{\ 0.54} \qquad (6.4)$$

C_{hw} = roughness coefficient associated with the pipe material

2) **Chezy**: open channel flow,

$$V = C\sqrt{R_h S} \quad \text{where} \quad C = \sqrt{\frac{8g_n}{f}}$$

A. Chezy (1718~1798; French)

TABLE 1 Hazen-Williams Coefficient C_{hw} and Manning n-values[a]

	C_{hw}	n
Extremely smooth pipes—PVC	150–160	0.009
Copper, aluminum tubing	150	0.010
Asbestos cement	140	0.011
New cast iron	130	0.013
Welded steel	130–140	0.012
Concrete	120–140	0.011–0.014
Ductile iron (cement lined)	140	0.011
Vitrified clay pipe	—	0.011–0.013
Riveted steel	110	0.013–0.017
Old cast iron	100	0.015–0.035

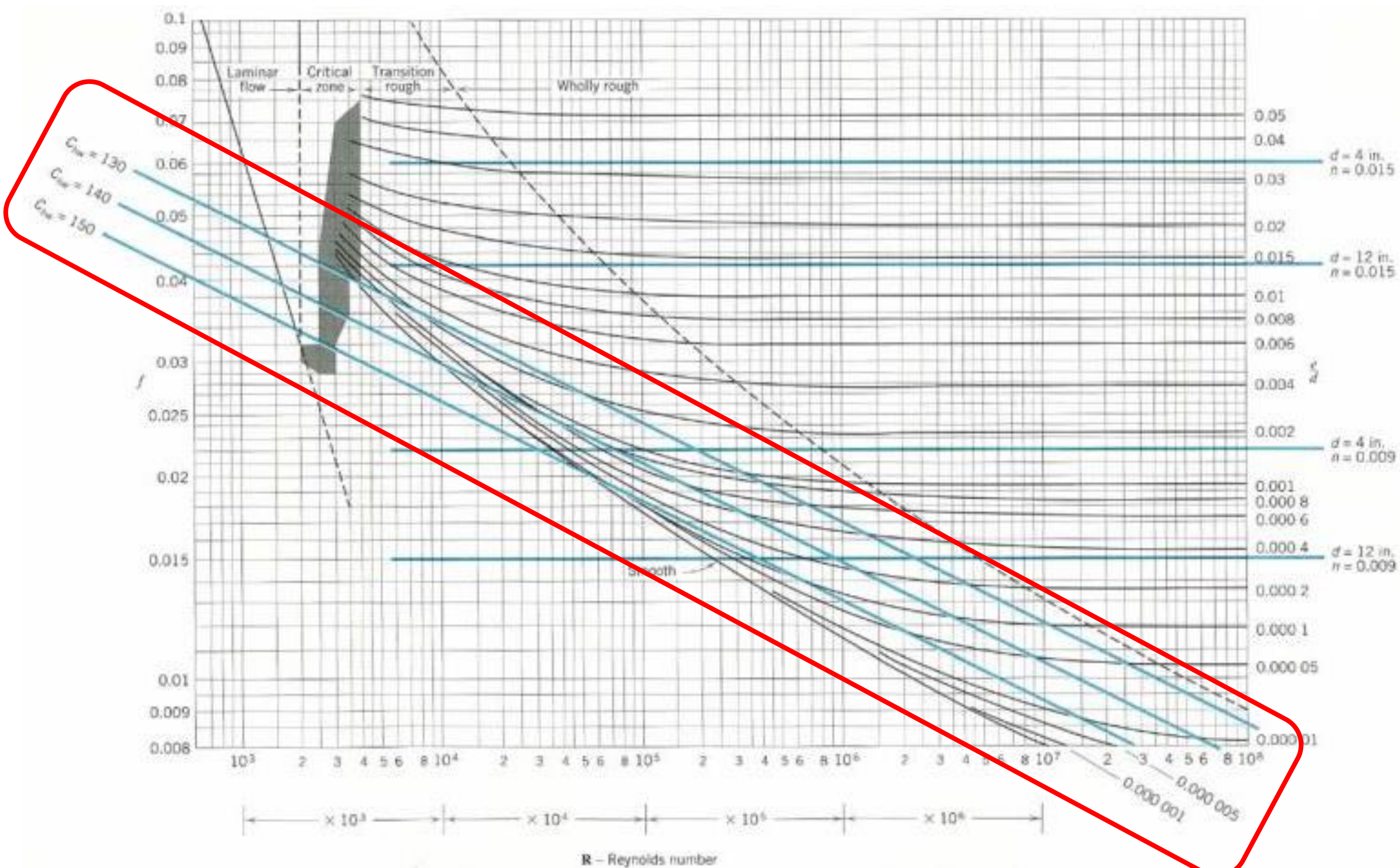
[a]These are typical values but, because of variabilities in fabrication, the user should consult the pipe manufacturer for recommended values of roughness coefficients.

- To judge the range of validity of Hazen-Williams formula, Eq. 6.4 is rewritten in the form of Darcy-Weisbach equation as

$$h_L = \left[\frac{116.3}{\nu^{0.15} C_{hw}^{\ 1.85} d^{0.015} \mathrm{Re}^{0.15}} \right] \frac{l}{d} \frac{V^2}{2g_n} = f' \frac{l}{d} \frac{V^2}{2g_n}$$

$$f' = \frac{923.4}{C_{hw}^{\ 1.85} d^{0.015} \mathrm{Re}^{0.15}}$$

- f' is plotted in the Moody diagram. → Figure 9.12 in p. 355
  - Hazen-Williams formula is a transition to smooth pipe formula.
  - There is a small relative roughness effect.
  - There is definitely a strong Reynolds number effect.


Laminar flow
Critical zone
Transition rough
Wholly rough
C_{hw} = 130
C_{hw} = 140
C_{hw} = 150
f
0.1
0.09
0.08
0.07
0.06
0.05
0.04
0.03
0.025
0.02
0.015
0.01
0.009
0.008
Smooth
0.05
0.04
0.03
0.02
0.015
0.01
0.008
0.006
0.004
0.002
0.001
0.000 8
0.000 6
0.000 4
0.000 2
0.000 1
0.000 05
0.000 01
0.000 001
0.000 005
e/d
d = 4 in. n = 0.015
d = 12 in. n = 0.015
d = 4 in. n = 0.009
d = 12 in. n = 0.009
10³ 2 3 4 5 6 8 10⁴ 2 3 4 5 6 8 10⁵ 2 3 4 5 6 8 10⁶ 2 3 4 5 6 8 10⁷ 2 3 4 5 6 8 10⁸
× 10³
× 10⁴
× 10⁵
× 10⁶
R – Reynolds number

R. Manning (1816~1897; Irish)

3) **Manning** equation

- Application to ***open channel***, but also used for pipe flow.

(SI units) $$V = \frac{1}{n} R_h^{2/3} S^{1/2} \qquad (6.5)$$

- To investigate the range of applicability of the Manning formula, arranging Eq. 6.5 in the Darcy-Weisbach equation (Fig. 9.12)

$$h_L = \left[\frac{124.4n^2}{d^{1/3}} \right] \frac{l}{d} \frac{V^2}{2g_n} = f'' \frac{l}{d} \frac{V^2}{2g_n}$$

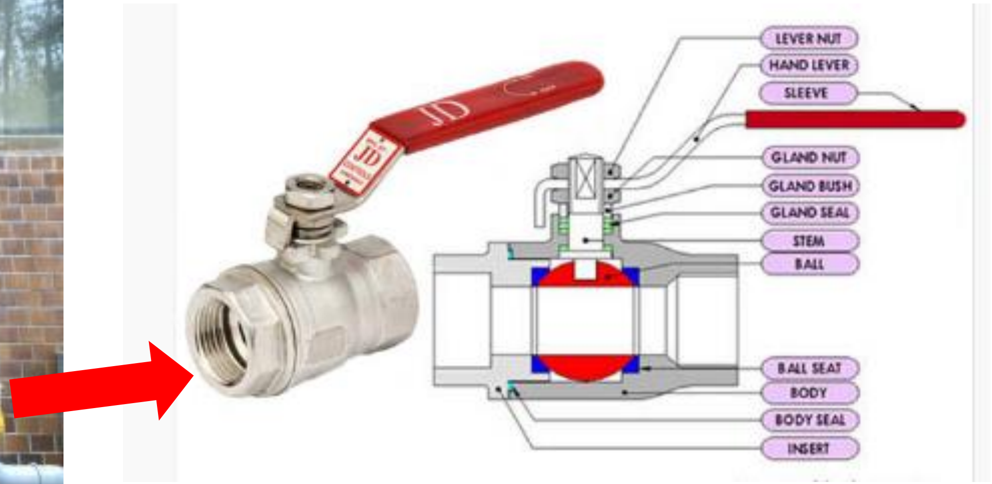
$$f'' = \frac{124.4n^2}{d^{1/3}}$$

- Manning's formula
  - There is no Reynolds number effect so the formula must be used only in the ***wholly rough flow zone*** where its horizontal slope can accurately match Darcy-Weisbach values provided the proper *n*-value is selected.
  - The relative roughness effect is correct in the sense that, for a given roughness, a ***larger pipe will have a smaller factor***.
  - In general sense, because the formula is ***valid only for rough pipes***, the rougher the pipe, the more likely the Manning formula will apply.

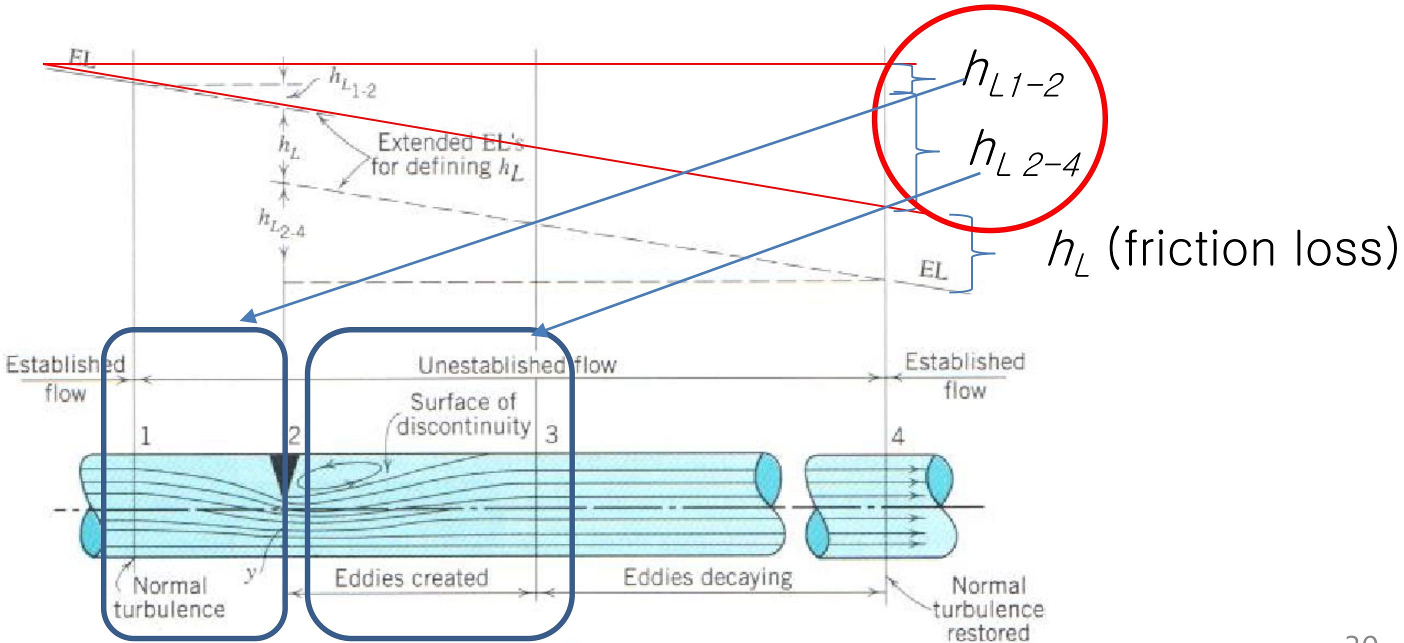
6.4 Local losses in pipelines

- Bends, elbow, valves, and fittings. Those have the change of cross-section and it causes the head loss.
- In the long pipes, such effects can be neglected but in the short ones, those are significant.
- For example, an abrupt obstruction placed in a pipeline creates dissipation of energy and causes local loss.
- This is generated by the velocity change mainly.
- Increase of velocity (acceleration) is associated with small head loss.
- But, decrease of velocity (deceleration) causes large head loss due to the large scale turbulence.

Ball valve


LEVER NUT
HAND LEVER
SLEEVE
GLAND NUT
GLAND BUSH
GLAND SEAL
STEM
BALL
BALL SEAT
BODY
BODY SEAL
INSERT

- Head loss
  - The useful energy is extracted to create eddies (large scale turbulence) as the fluid decelerates between 2 and 3.
  - And eddies decay at Sec. 3-4 by dissipation to heat.
  - Therefore, the local loss in pipe flow is accomplished in the pipe downstream from the source of large scale eddy (or turbulence).



- Earlier experiments with water (at high Reynolds number) indicated that local losses vary approximately with the square of velocity and led to the proposal of the basic equation.

$$h_L = K_L \frac{V^2}{2g_n} \qquad K_L \text{ is the loss coefficient} \qquad (6.6)$$

- Loss coefficient

- Tends to increase with increasing roughness

- Increases with decreasing Reynolds number

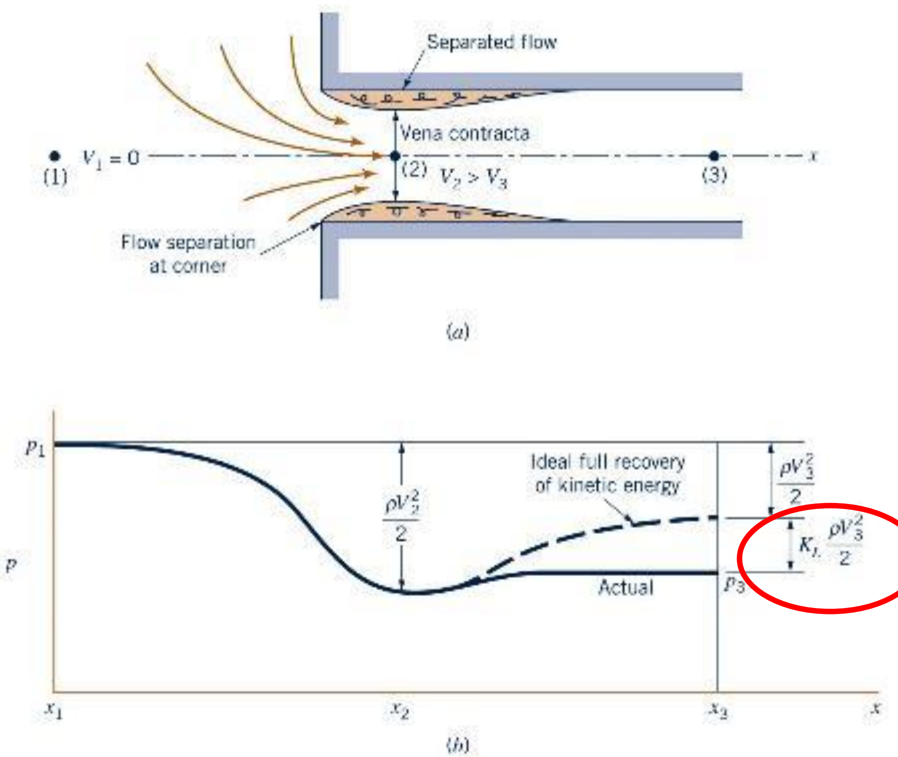
- Constant at the real high Reynolds number (wholly turbulent flow)

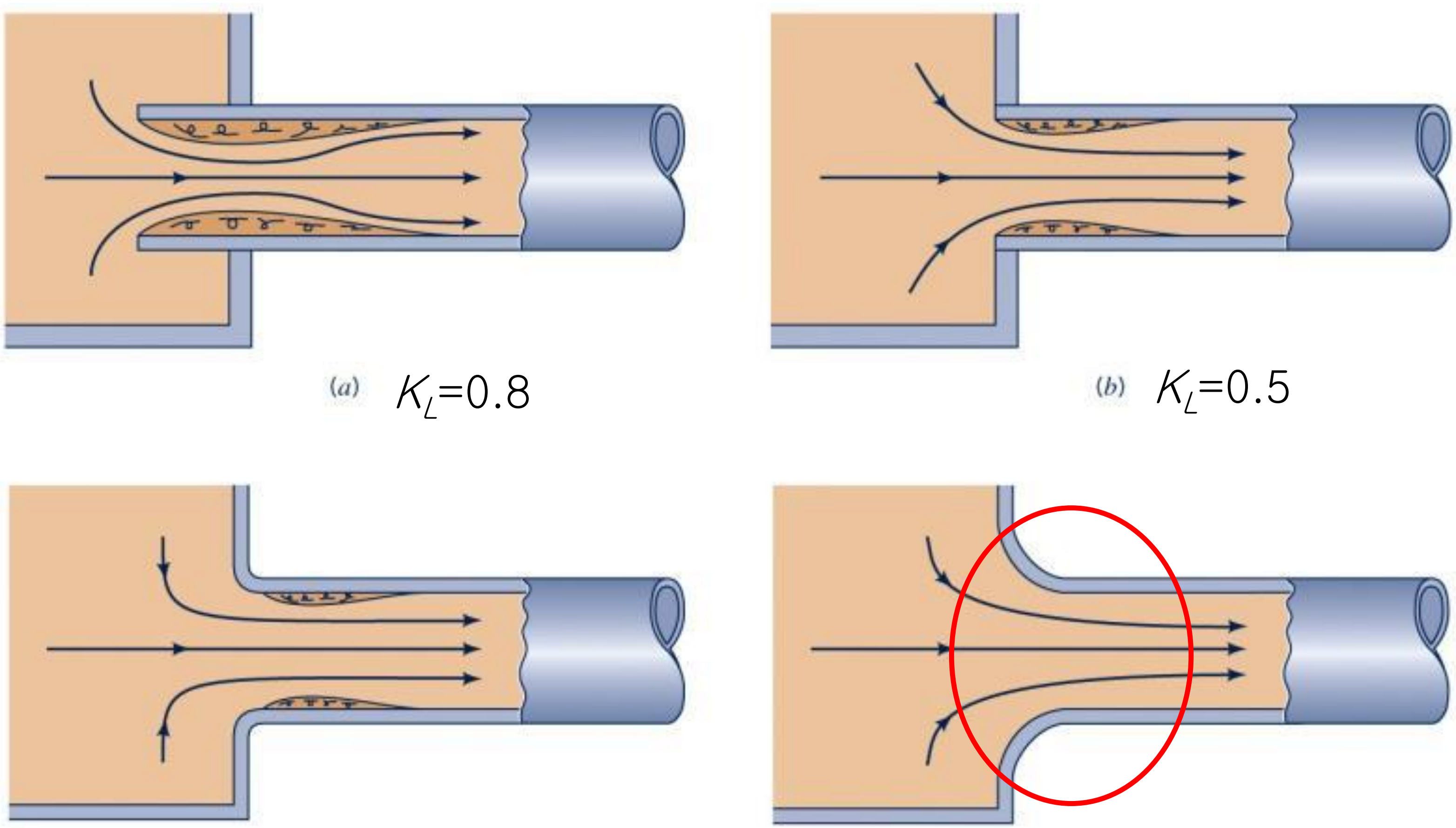
- Mainly determined by the geometry and the shape of the obstruction or pipe fitting.

[Cf] Friction loss: $h_L = f \frac{l}{d} \frac{V^2}{2g_n}$

I. Entrances

- Square edge
  - Vena contracta region may result because the fluid cannot turn a sharp right-angle corner.
    - At Vena contracta region, a fluid may be accelerated very efficiently, and pressure there is low.
    - However, at region (3), it is very difficult to decelerate a fluid efficiently.
    - Thus, kinetic energy at (2) is partially lost because of viscous dissipation.





(a) K_L=0.8

(b) K_L=0.5

(c) K_L=0.2

(d) K_L=0.04

Vena contracta not exists

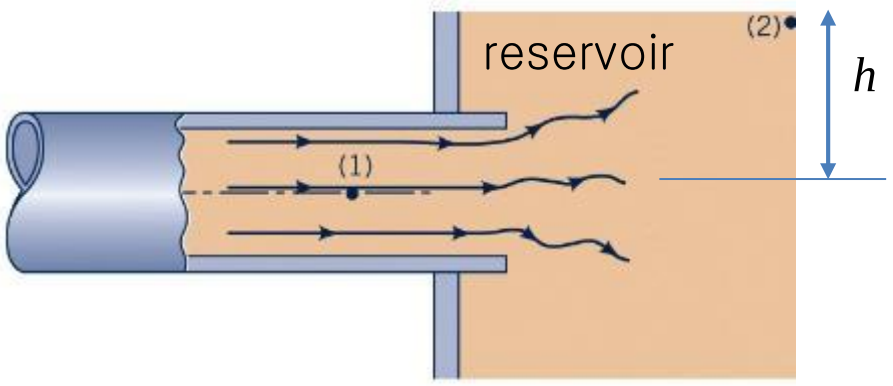
II. Exit

- Exit loss
  - Kinetic energy of the exiting fluid (V_1) is dissipated through viscous effects as the stream of fluid mixed with the fluid in the reservoir or tank and eventually comes to rest (V_2 = 0).

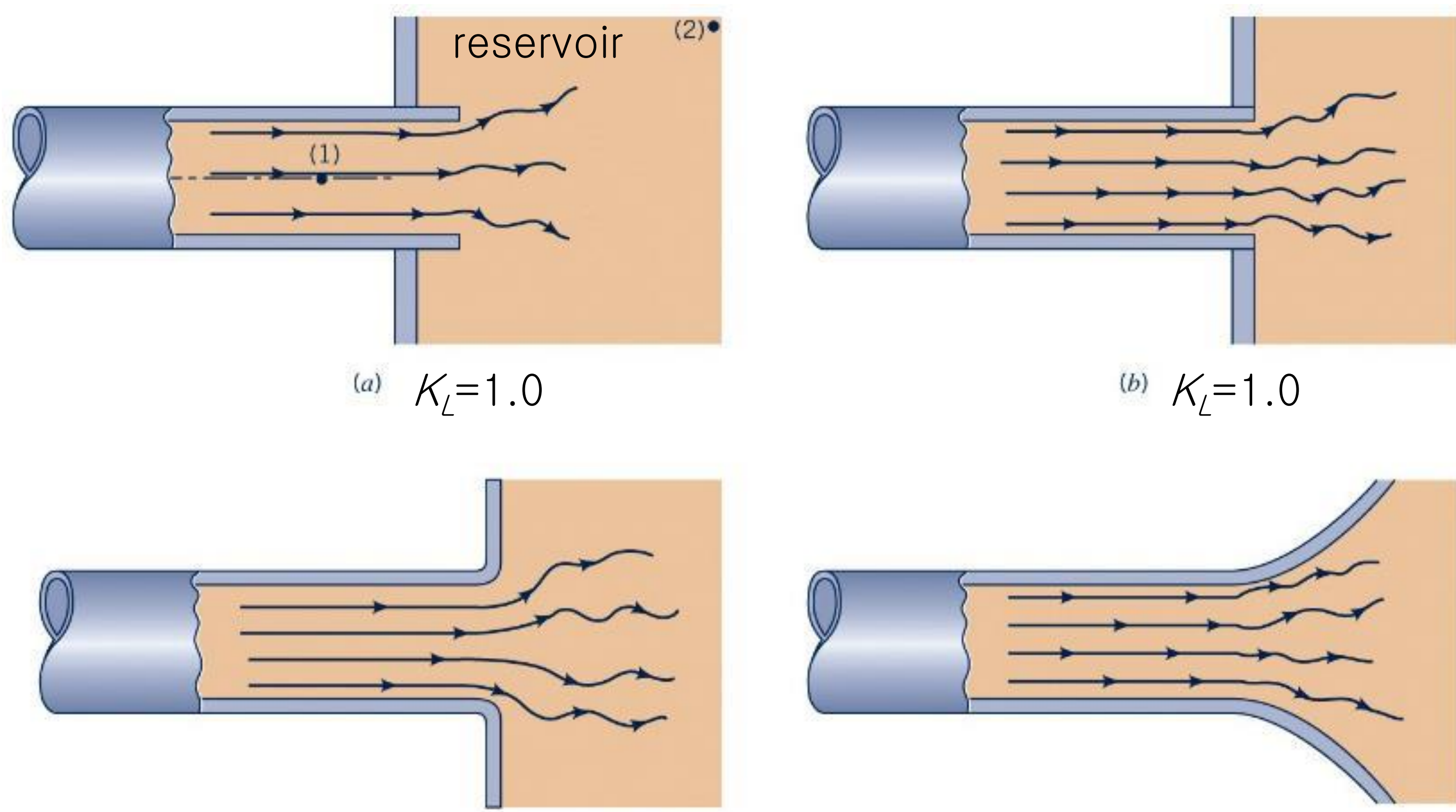
$$z_1 + \frac{p_1}{\gamma} + \frac{V_1^2}{2g_n} = z_2 + \frac{p_2}{\gamma} + \frac{V_2^2}{2g_n} + h_L$$

$$h_L = \frac{V_1^2}{2g_n} + (z_1 - z_2) + \frac{p_1}{\gamma}$$

$$= \frac{V_1^2}{2g_n} - h + \frac{p_1}{\gamma} = \frac{V_1^2}{2g_n}$$



Exit loss



(a) K_L=1.0

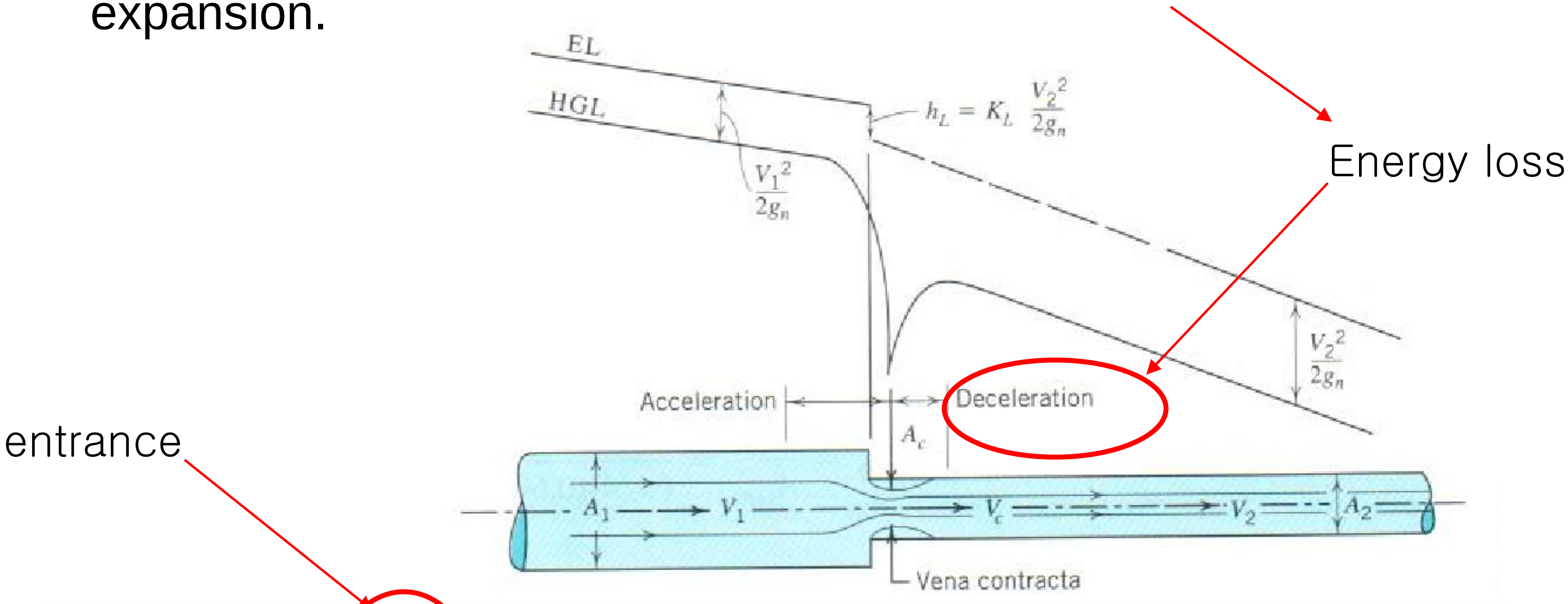
(b) K_L=1.0

(c) K_L=1.0

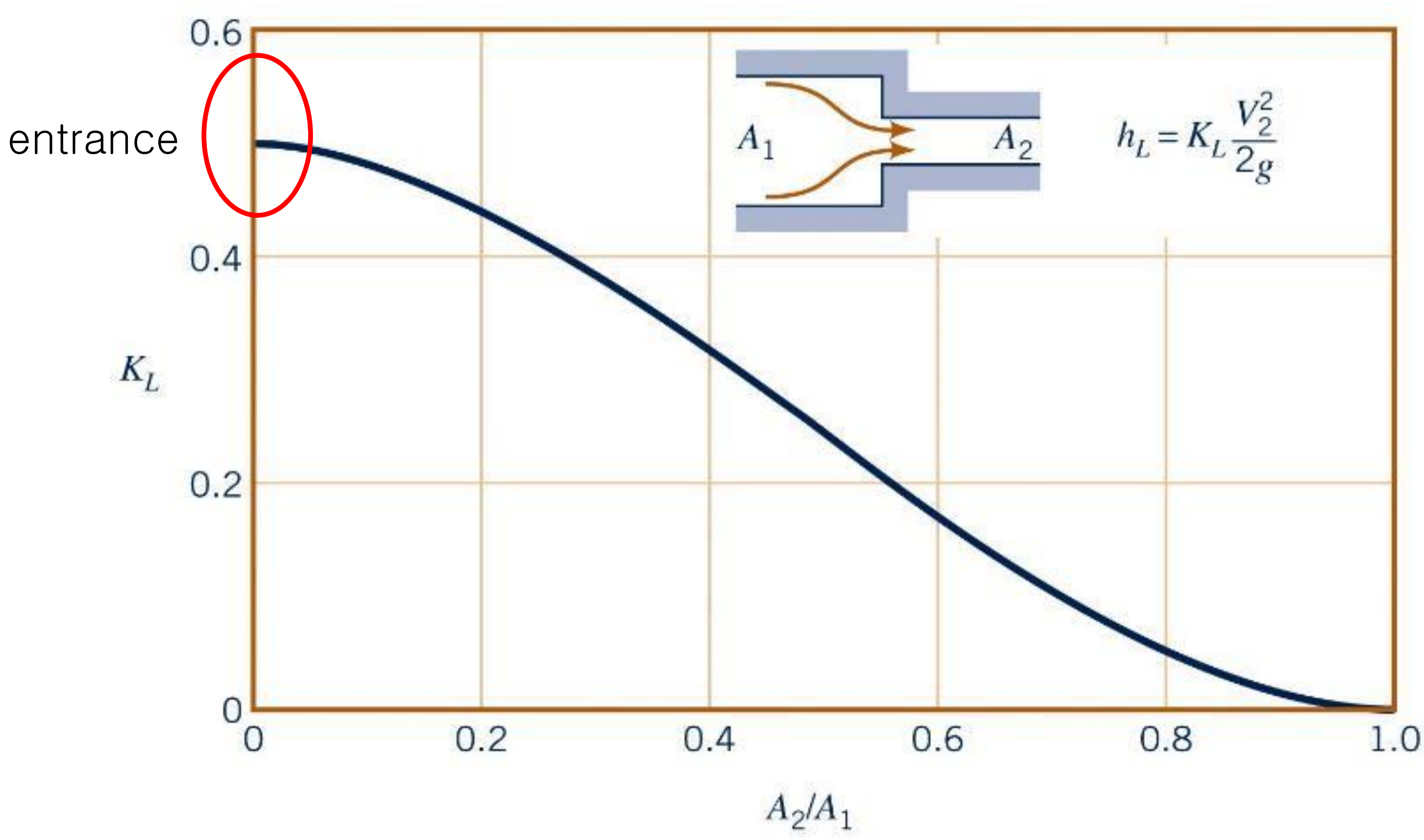
(d) K_L=1.0

III. Contractions

- Flow through an ***abrupt contraction*** and is featured by the formation of a Vena contracta and subsequent deceleration and re-expansion.



A_2/A_1	0	0.2	0.4	0.6	0.8	1.0
$C_c=A_c/A_2$	0.617	0.632	0.659	0.712	0.813	1.00
K_L	0.50	0.41	0.30	0.18	0.06	0

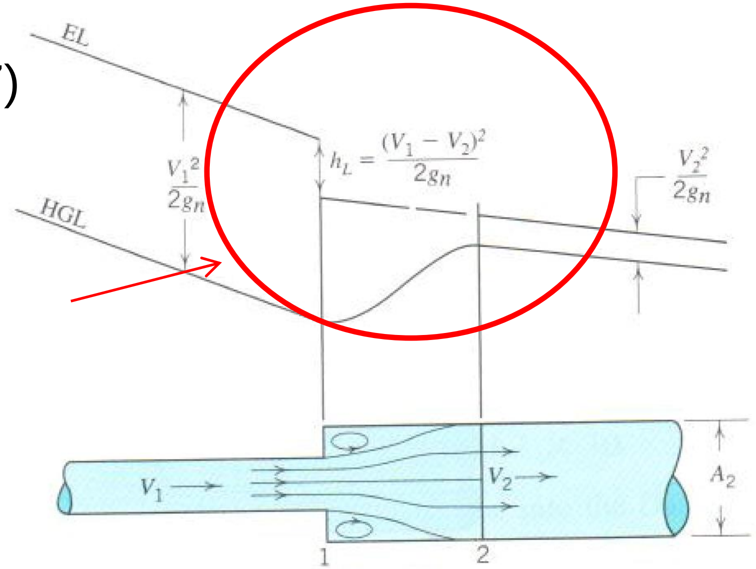

entrance
0.6
0.4
0.2
0
K_L
0
0.2
0.4
0.6
0.8
1.0
A_2/A_1
A_1
A_2
$h_L = K_L \frac{V_2^2}{2g}$

IV. Enlargement (Expansion)

- When an ***abrupt enlargement*** of section occurs in a pipeline, a rapid deceleration takes place, accompanied by characteristic large-scale turbulence
- It will persist for a distance of 50 diameter or more down stream.

$$h_L = K_L \frac{(V_1 - V_2)^2}{2g_n} \qquad K_L \cong 1 \qquad (6.7)$$

Compare the slopes of EL



[Re] Abrupt enlargement in a closed passage ~ real fluid flow

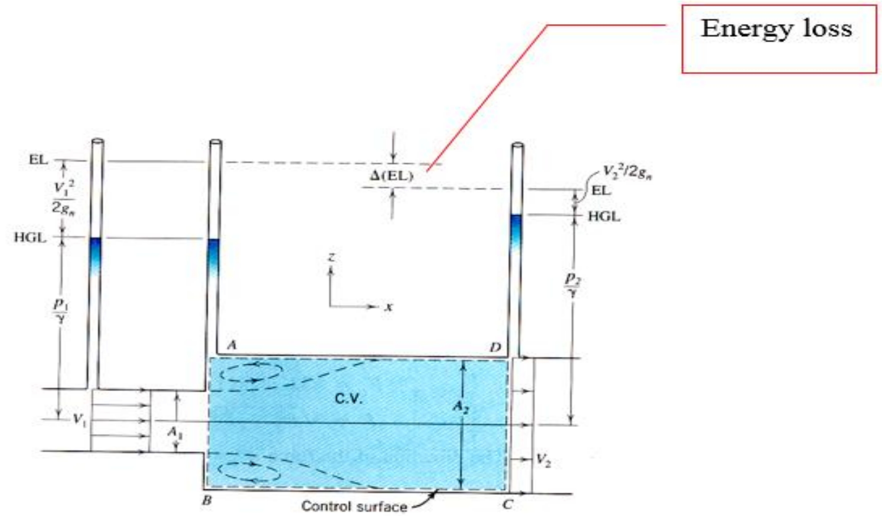
The impulse-momentum principle can be employed to predict the fall of the energy line (energy loss due to a rise in the internal energy of the fluid caused by viscous dissipation due to eddy formation) at an abrupt axisymmetric enlargement in a passage.

Neglect friction force at the pipe wall.

Consider the control surface *ABCD* assuming a one-dimensional flow

i) Continuity Eq.

$$Q = A_1 V_1 = A_2 V_2$$



ii) Momentum Eq.

Result from hydrostatic pressure distribution over the area

→ For area *AB* it is an approximation because of the dynamics of eddies in the “dead water” zone.

$$\sum F_x = p_1 A_2 - p_2 A_2 = Q\rho (V_2 - V_1)$$

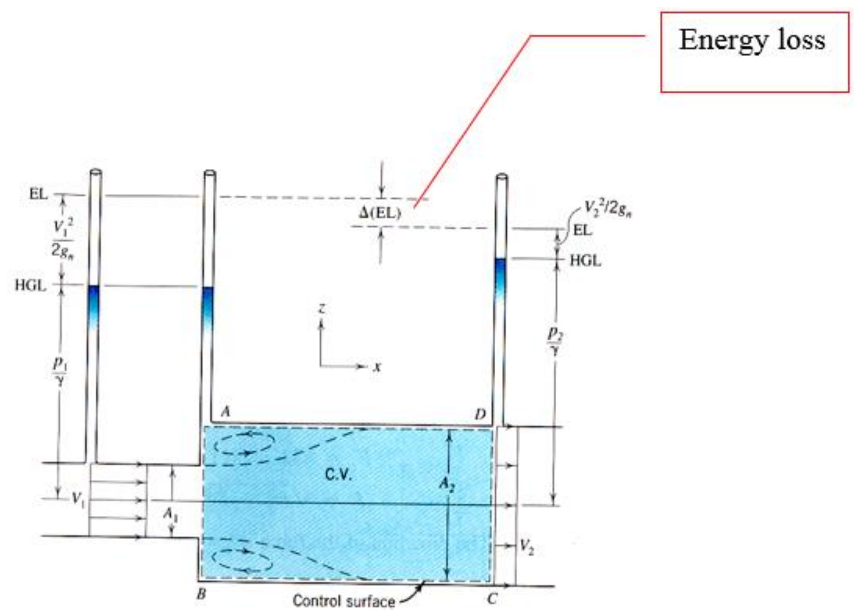
$$(p_1 - p_2) A_2 = \frac{V_2 A_2}{g} \gamma (V_2 - V_1)$$

$$\therefore \; \frac{p_1 - p_2}{\gamma} = \frac{V_2}{g}(V_2 - V_1) \qquad \text{(a)}$$

iii) Bernoulli Eq.

$$\frac{p_1}{\gamma} + \frac{V_1^2}{2g} = \frac{p_2}{\gamma} + \frac{V_2^2}{2g} + h_L$$

$$\frac{p_1 - p_2}{\gamma} = \frac{V_2^2}{2g} - \frac{V_1^2}{2g} + h_L \qquad \text{(b)}$$



where $h_L =$ Borda−Carnot head loss

Jean-Charles de Borda (1733~1799): French mathematician

Nicolas Leonard Sadi Carnot (1796~1832): French military engineer

Combine (a) and (b)

$$\frac{V_2(V_2 - V_1)}{g} = \frac{V_2^2}{2g} - \frac{V_1^2}{2g} + h_L$$

$$h_L = \frac{2V_2^2 - 2V_1V_2}{2g} - \frac{V_2^2}{2g} + \frac{V_1^2}{2g} = \boxed{\frac{(V_1 - V_2)^2}{2g}} \qquad (6.8)$$

[Re] Conversion

– change formula in terms of V_1

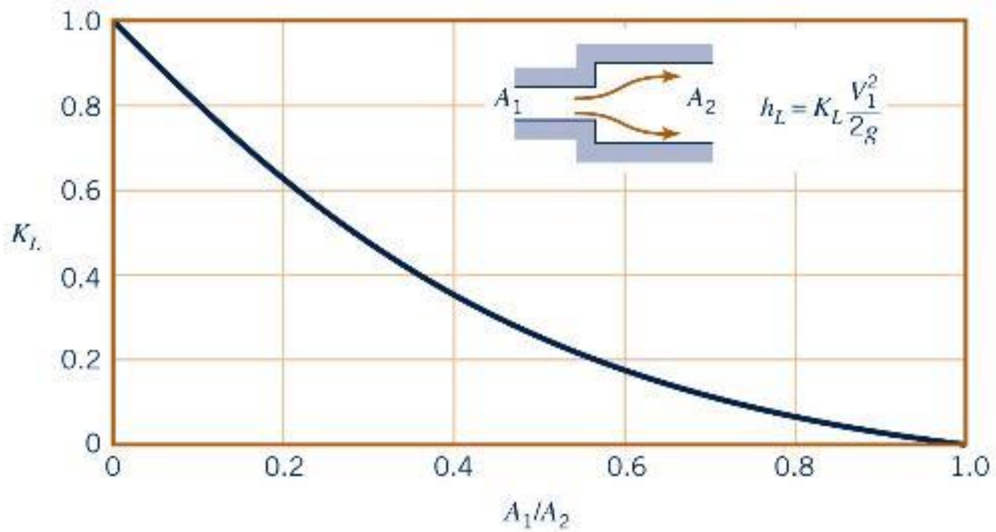
$$h_L = K_L \frac{(V_1 - V_2)^2}{2g}$$

$$Q = A_1 V_1 = A_2 V_2$$

– Combine two equations

$$h_L = K_L{}' \frac{V_1^2}{2g}$$

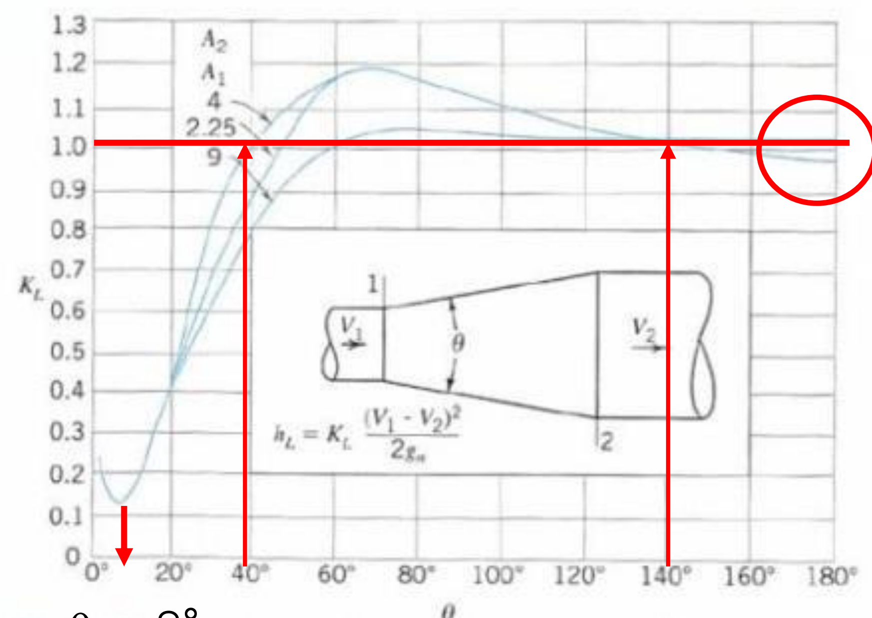
$$K_L{}' = \left(1 - \frac{A_1}{A_2}\right)^2 K_L$$



V. Gradual enlargement

$$h_L = K_L \frac{(V_1 - V_2)^2}{2g} \qquad (6.9)$$

- The loss of head due to ***gradual enlargement*** is dependent on the shape of the enlargement. → central angle and area ratio (A_2/A_1)
  - In an enlargement of small central angle, effects of wall friction should be accounted for as well effect of large-scale turbulence.
  - For moderate angle (θ = 40°~140°), K_L is larger than 1, which means that these diffuser is less efficient than the sharp-edged expansion.



θ = 8°

I.P 9.13 (p. 361)

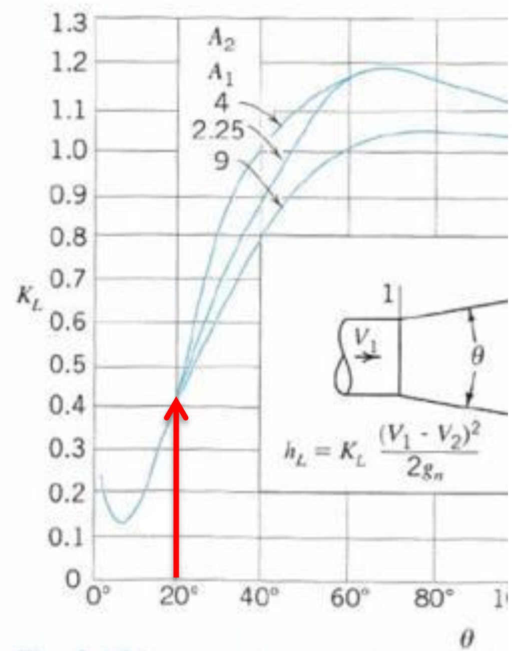
- A 300 mm horizontal water line enlarges to a 600 mm line through 20° conical enlargement. When 0.30 m^3/s flow through this line, the pressure in the smaller pipe is 140 kPa. Calculate the pressure in the larger pipe, neglecting pipe friction.

- Velocities in each pipe

$$V_{300} = \frac{Q}{A_{300}} = \frac{0.30 m^3/s}{(p/4)(0.300 m)^2} = 4.24 m/s$$

$$V_{600} = \frac{Q}{A_{600}} = \frac{0.30 m^3/s}{(p/4)(0.600 m)^2} = 1.06 m/s$$

- K_L = 0.43 (see the figure; A_2/A_1 = 4; θ = 20°)



$$h_L = K_L \frac{\left(V_{300} - V_{600}\right)^2}{2g_n} = 0.43\frac{\left(4.24 - 1.06\right)^2}{2 \times 9.81} = 0.222\,m$$

- To compute the pressure in the large pipe, use Bernoulli's eq.

$$z_{300} + \frac{p_{300}}{\gamma} + \frac{V_{300}^2}{2g_n} = z_{600} + \frac{p_{600}}{\gamma} + \frac{V_{600}^2}{2g_n} + h_L$$

- Taking the datum as the pipe centerline eliminates *z* from the calculations

$$\frac{140 \times 10^3\,\text{Pa}}{9{,}800\,\text{N/m}^3} + \frac{\left(4.24\text{m/s}\right)^2}{2 \times 9.81} = \frac{p_{600}}{\gamma} + \frac{\left(1.06\text{m/s}\right)^2}{2 \times 9.81} + 0.222$$

$$\frac{p_{600}}{\gamma} = 14.6\text{m}, \qquad p_{600} = 14.6 \times 9{,}800 = 143\text{kPa}$$

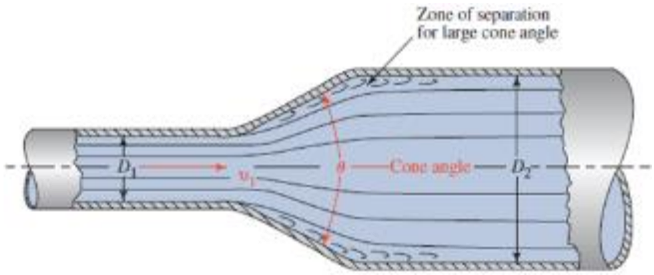
- Pressure increases at 2 compared to 1.

VI. Gradual contraction

For gradual contraction, K_L is smaller than the gradual enlargements.

$$K_L = 0.02 \text{ for } \theta = 30^{\circ}$$

$$K_L = 0.07 \text{ for } \theta = 60^{\circ}$$



- Increase of velocity (acceleration) is associated with small head loss.

- Decrease of velocity (deceleration) causes large head loss due to the large scale turbulence.

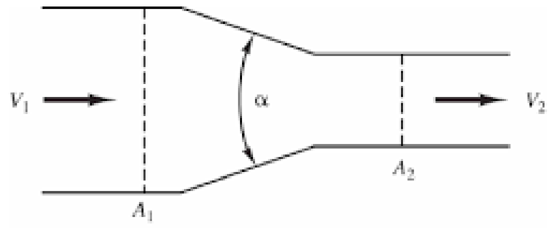


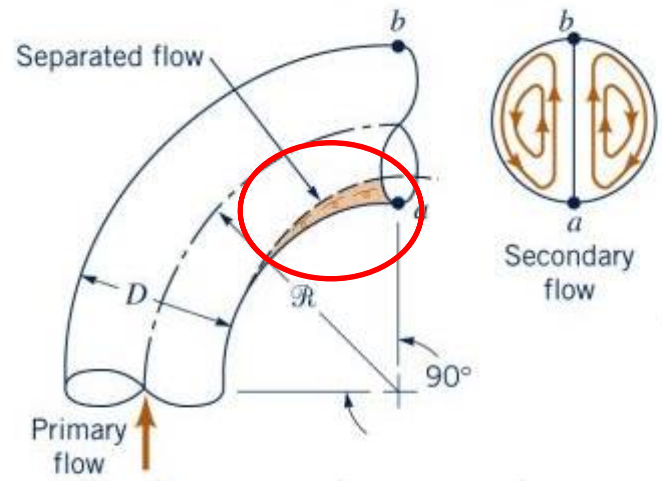
Figure 2.2. Pipe Confusor

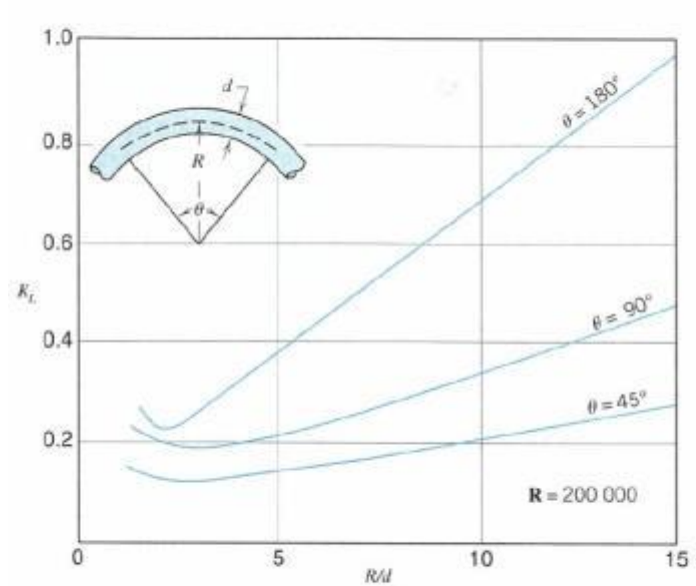
VII. Bends

- Losses of head in ***smooth pipe bends*** are caused by the combined effects of separation, wall friction and the twin-eddy secondary flow (Fig. 7.29 in p. 274).
- For bends of large radius of curvature, the last two effects will predominate,
- For small radius of curvature, sharp bend, separation and the secondary flow will be the more significant.

$$h_L = K_L \frac{V^2}{2g_n} \qquad (6.10)$$

- K_L is a function of θ, *R/d, and Reynolds number*.



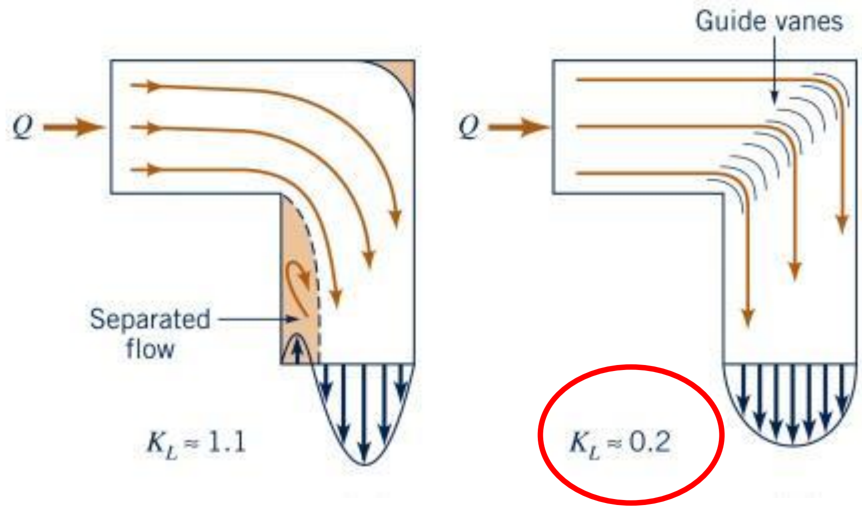


VIII. Miter Bends

- ***Miter bends*** are used in large ducts where space does not permit a bend of large radius.

$$K_L \sim 1.1$$

- Installation of guide vanes reduces the head loss and breaks up the spiral motion and improve the velocity distribution downstream.



IX. Commercial pipe fittings

- The head losses caused by ***commercial pipe fittings*** occur because of their rough and irregular shapes which produce excessively large-scale turbulence.

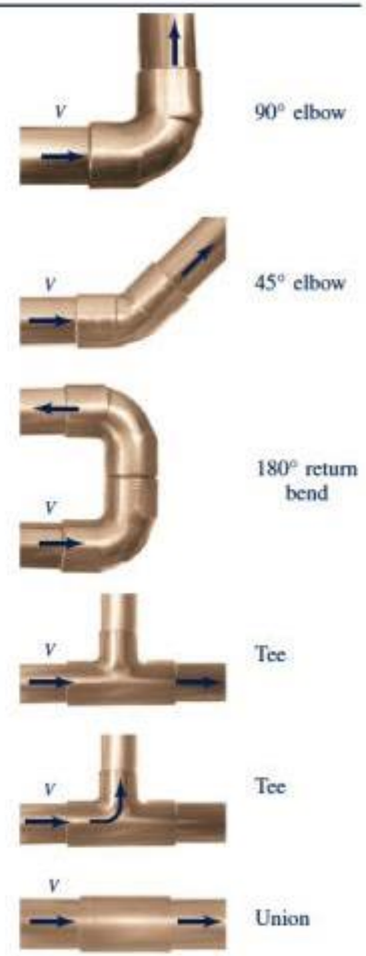
$$K_L \sim Engineering\ Data\ Book\ (Table\ 3)$$

TABLE 3 Approximate Loss Coefficient, K_L, for Commercial Pipe Fittings[21]

<table>
<tr><th>Valves, wide open</th><th>Screwed</th><th>Flanged</th></tr>
<tr><td>Globe</td><td>10</td><td>5</td></tr>
<tr><td>Gate</td><td>0.2</td><td>0.1</td></tr>
<tr><td>Swing-check</td><td colspan="2">2</td></tr>
<tr><td>Angle</td><td colspan="2">2</td></tr>
<tr><td>Foot</td><td colspan="2">0.8</td></tr>
<tr><td>Return bend</td><td>1.5</td><td>0.2</td></tr>
<tr><td>Elbows</td><td></td><td></td></tr>
<tr><td>90°—regular</td><td>1.5</td><td>0.3</td></tr>
<tr><td>—long radius</td><td>0.7</td><td>0.2</td></tr>
<tr><td>45°—regular</td><td>0.4</td><td>—</td></tr>
<tr><td>—long radius</td><td>—</td><td>0.2</td></tr>
<tr><td>Tees</td><td></td><td></td></tr>
<tr><td>Line flow</td><td>0.9</td><td>0.2</td></tr>
<tr><td>Branch flow</td><td>2</td><td>1</td></tr>
</table>

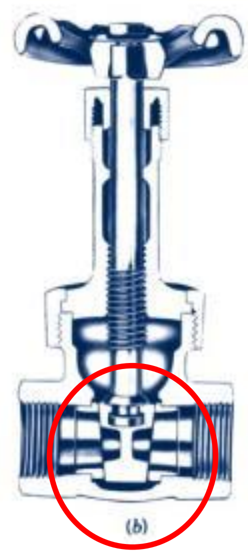
screwed

Component	K_L
a. Elbows	
Regular 90°, flanged	0.3
Regular 90°, threaded	1.5
Long radius 90°, flanged	0.2
Long radius 90°, threaded	0.7
Long radius 45°, flanged	0.2
Regular 45°, threaded	0.4
b. 180° return bends	
180° return bend, flanged	0.2
180° return bend, threaded	1.5
c. Tees	
Line flow, flanged	0.2
Line flow, threaded	0.9
Branch flow, flanged	1.0
Branch flow, threaded	2.0
d. Union, threaded	0.08
***e. Valves**	
Globe, fully open	10
Angle, fully open	2
Gate, fully open	0.15
Gate, $\frac{1}{4}$ closed	0.26
Gate, $\frac{1}{2}$ closed	2.1
Gate, $\frac{3}{4}$ closed	17
Swing check, forward flow	2
Swing check, backward flow	∞
Ball valve, fully open	0.05
Ball valve, $\frac{1}{3}$ closed	5.5
Ball valve, $\frac{2}{3}$ closed	210

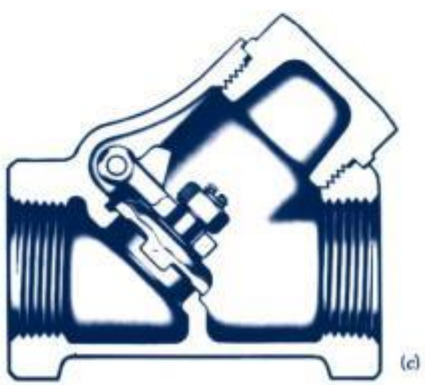


Globe valve

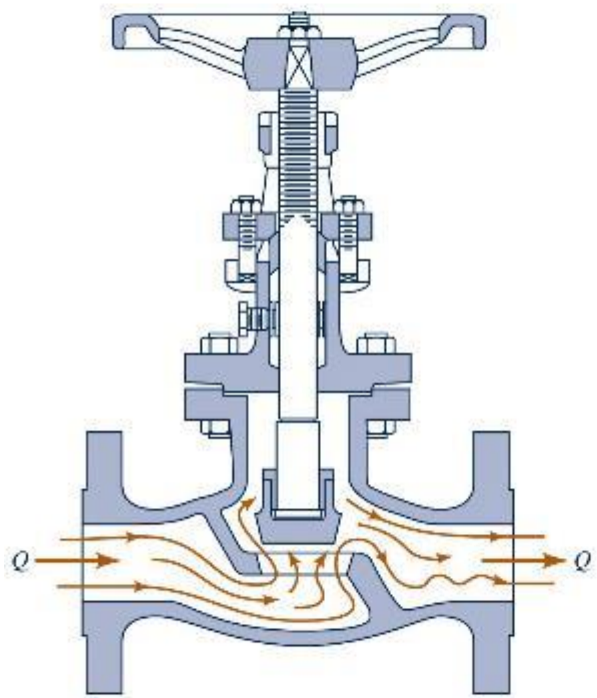
Gate valve



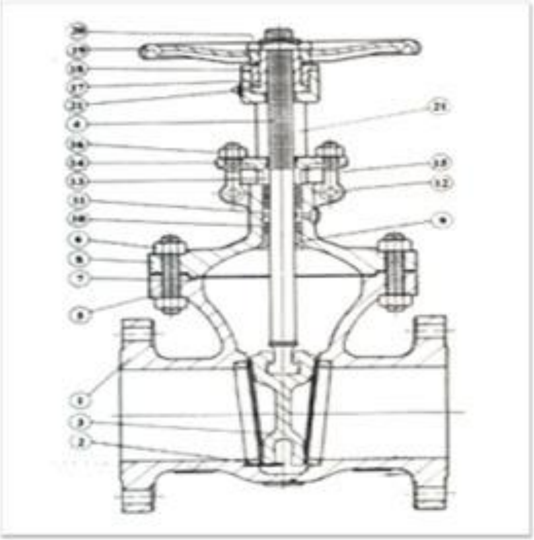
Swing check valve



Stop check valve



Globe valve



Gate valve

Homework Assignment No. 3
Due: 2 weeks from today
Answer questions in Korean or English

1. (9-1) When 0.3 m^3/s of water flows through a 150 *mm* constriction in a 300 *mm* horizontal pipeline, the pressure at a point in the pipe is 345 *kPa*, and the head lost between this point and the constriction is 3 *m*. Calculate the pressure in the constriction.

2. (9-6) A pump of what power is required to pump 0.56 m^3/s of water from a reservoir of surface elevation 30 to the reservoir of surface elevation 75, if in the pump and pipeline 12 meters of head are lost?

3. (9-11) When a horizontal laminar flow occurs between two parallel plates if infinite extent 0.3 *m* apart, the velocity at the midpoint between the plates is 2.7 m^3/s . Calculate (*a*) the flowrate through a cross section 0.9 *m* wide, (*b*) the velocity gradient at the surface of the plate, (*c*) the wall shearing stress if the fluid has viscosity 1.44 *Pa.s*, (*d*) the pressure drop in each 30 *m* along the flow.

4. (9-19) In a turbulent flow in a 0.3 *m* pipe the centerline velocity is 6 *m/s*, and that 50 *mm* from the pipe wall 5.2 *m/s*. Calculate the friction factor and flowrate.

5. (9-35) Solve Problem 3 for turbulent flow, rough plates with *e* = 0.5 *mm*, and fluid density and viscosity 1000 kg/m^3 and 0.0014 *Pa.s,* respectively.

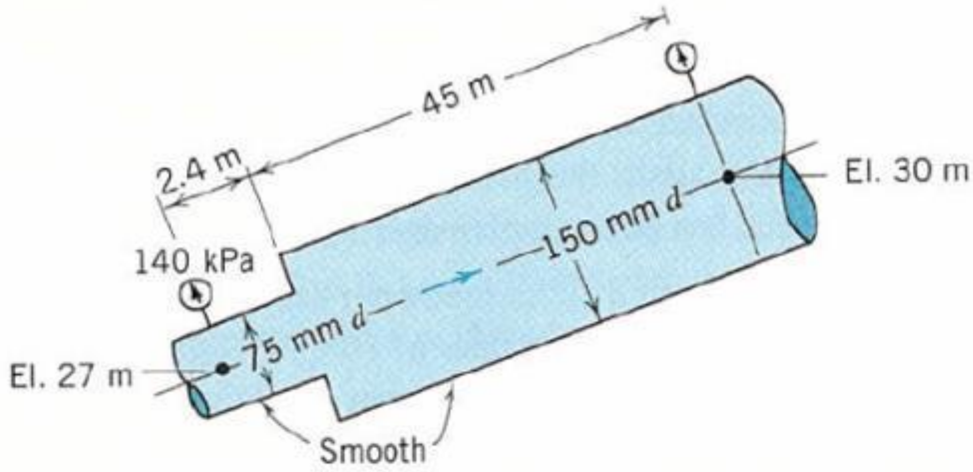
6. (9-44) A single layer of steel spheres is stuck to the glass-smooth floor of a two-dimensional open channel. Water of kinematic viscosity 9.3 x 10^{-7} m^2/s flows in the channel at a depth of 0.3 *m* the surface velocity of $\frac{1}{4}$ *m/s*. Show that for spheres of 7.2 *mm* and 0.3 *mm* diameter that the channel bottom should be classified *rough* and *smooth*, respectively.

7. (9-55) Calculate the loss of head in 300 *m* of 75 *mm* PVC pipe when water at 27 ℃ flows therein at a mean velocity of 3 *m/s* .

8. (9-58) If 0.34 m^3/s of water flows in a 0.3 *m* riveted steel pipe at 21 ℃, calculate the smallest loss of head to be expected on 150 *m* of the pipe.

9. (9-78) Three-tenths of a cubic meter per second of water flows in a smooth 230 mm square duct at 10 ℃. Calculate the head lost in 30 *m* of this duct.

10. (9–89) The fluid flowing has specific gravity 0.90; V_{75} = 6 *m/s*; *Re* = 10^5. Calculate the gage reading.



11. (9–106) A 90° screwed elbow is installed in a 50 *mm* pipeline having a friction factor of 0.03. The head lost at the elbow is equivalent to that lost in how many meters of the pipe? Repeat the calculation for a 25 *mm* pipe.