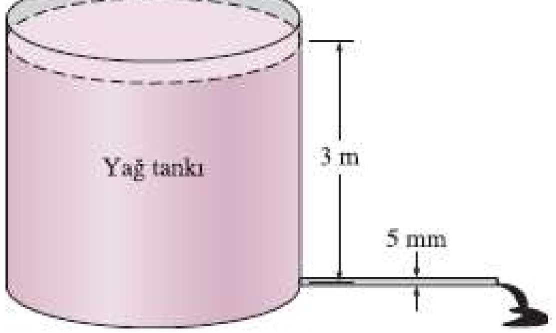
8–30 Yoğunluğu 850 kg/m^3 ve kinematik viskozitesi 0.00062 m^2/s olan yağ, çapı 5 mm ve uzunluğu 40 m olan yatay boru ile atmosfere açık bir depolama tankından boşaltılmaktadır. Sıvı seviyesinin boru merkezinden yüksekliği 3 m'dir. Yerel kayıpları göz ardı ederek borudaki yağın debisini hesaplayınız.



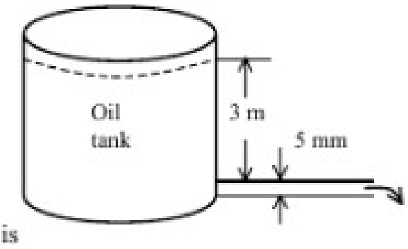
8-30 Oil is being discharged by a horizontal pipe from a storage tank open to the atmosphere. The flow rate of oil through the pipe is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The entrance and exit loses are negligible. **4** The flow is laminar (to be verified). **5** The pipe involves no components such as bends, valves, and connectors. **6** The piping section involves no work devices such as pumps and turbines.

Properties The density and kinematic viscosity of oil are given to be ρ = 850 kg/m^3 and ν = 0.00062 m^2/s, respectively. The dynamic viscosity is calculated to be

$$\mu = \rho \nu = (850\ \text{kg/m}^3)(0.00062\ \text{m}^2/s) = 0.527\ \text{kg/m}\cdot\text{s}$$

Analysis The pressure at the bottom of the tank is



$$P_{1,\text{gage}} = \rho g h$$

$$= (850\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(3\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)$$

$$= 25.02\ \text{kN/m}^2$$

Disregarding inlet and outlet losses, the pressure drop across the pipe is

$$\Delta P = P_1 - P_2 = P_1 - P_{\text{atm}} = P_{1,\text{gage}} = 25.02\ \text{kN/m}^2 = 25.02\ \text{kPa}$$

The flow rate through a horizontal pipe in laminar flow is determined from

$$\dot{V}_{\text{horiz}} = \frac{\Delta P \pi D^4}{128 \mu L} = \frac{(25.02\ \text{kN/m}^2)\pi(0.005\ \text{m})^4}{128(0.527\ \text{kg/m}\cdot\text{s})(40\ \text{m})}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right) = \mathbf{1.821\times10^{-8}\ m^3/s}$$

The average fluid velocity and the Reynolds number in this case are

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{1.821\times10^{-8}\ \text{m}^3/\text{s}}{\pi(0.005\ \text{m})^2/4} = 9.27\times10^{-4}\ \text{m/s}$$

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(850\ \text{kg/m}^3)(9.27\times10^{-4}\ \text{m/s})(0.005\ \text{m})}{0.527\ \text{kg/m}\cdot\text{s}} = 0.0075$$

which is less than 2300. Therefore, the flow is *laminar* and the analysis above is valid.

Discussion The flow rate will be even less when the inlet and outlet losses are considered, especially when the inlet is not well-rounded.

8–31 Sıcaklığı 10°C (ρ = 999.7 kg/m^3 ve μ = 1.307 × 10^{-3} kg/m · s) olan su, çapı 0.20 cm uzunluğu 15 m olan bir boruda 1.2 m/s'lik ortalama hız ile daimi olarak akmaktadır. (*a*) Basınç düşüşünü, (*b*) yük kaybını ve (*c*) bu basınç kaybını karşılamak için gereken pompalama gücünü bulunuz. *Cevaplar:* (*a*) 188 kPa, (*b*) 19.2 m, (*c*) 0.71 W

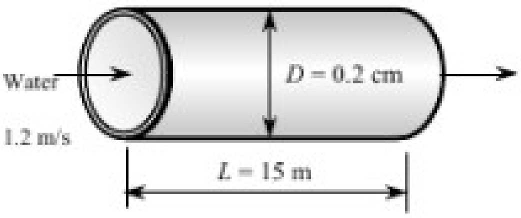
8-31 The average flow velocity in a pipe is given. The pressure drop, the head loss, and the pumping power are to be determined. √

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The pipe involves no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water are given to be ρ = 999.7 kg/m^3 and μ = 1.307×10^{-3} kg/m-s, respectively.

Analysis (*a*) First we need to determine the flow regime. The Reynolds number of the flow is

$$\mathrm{Re} = \frac{\rho VD}{\mu} = \frac{(999.7\ \mathrm{kg/m^3})(1.2\ \mathrm{m/s})(2\times10^{-3}\ \mathrm{m})}{1.307\times10^{-3}\ \mathrm{kg/m\cdot s}} = 1836$$



which is less than 2300. Therefore, the flow is laminar. Then the friction factor and the pressure drop become

$$f = \frac{64}{\mathrm{Re}} = \frac{64}{1836} = 0.0349$$

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.0349\frac{15\ \mathrm{m}}{0.002\ \mathrm{m}}\frac{(999.7\ \mathrm{kg/m^3})(1.2\ \mathrm{m/s})^2}{2}\left(\frac{1\ \mathrm{kN}}{1000\ \mathrm{kg\cdot m/s}}\right)\left(\frac{1\ \mathrm{kPa}}{1\ \mathrm{kN/m^2}}\right) = \mathbf{188\ kPa}$$

(*b*) The head loss in the pipe is determined from

$$h_L = \frac{\Delta P_L}{\rho g} = f\frac{L}{D}\frac{V^2}{2g} = 0.0349\frac{15\ \mathrm{m}}{0.002\ \mathrm{m}}\frac{(1.2\ \mathrm{m/s})^2}{2(9.81\ \mathrm{m/s^2})} = \mathbf{19.2\ m}$$

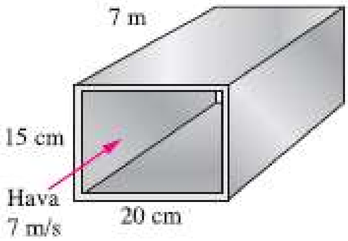
(*c*) The volume flow rate and the pumping power requirements are

$$\dot{V} = VA_c = V(\pi D^2/4) = (1.2\ \mathrm{m/s})[\pi(0.002\ \mathrm{m})^2/4] = 3.77\times10^{-6}\ \mathrm{m^3}/s$$

$$\dot{W}_{\mathrm{pump}} = \dot{V}\Delta P = (3.77\times10^{-6}\ \mathrm{m^3}/s)(188\ \mathrm{kPa})\left(\frac{1000\ \mathrm{W}}{1\ \mathrm{kPa\cdot m^3/s}}\right) = \mathbf{0.71\ W}$$

Therefore, power input in the amount of 0.71 W is needed to overcome the frictional losses in the flow due to viscosity.

8–41 Ticari çelikten yapılmış dikdörtgen en-kesitli (15 cm × 20 cm) bir kanalın, uzunluğu 7 m olan bölümünde 1 atm basınçta 35°C sıcaklıkta ve ortalama hızı 7 m/s olan hava akmaktadır. Giriş etkilerini göz ardı ederek kanalın bu bölümündeki basınç kaybını karşılamak için gereken fan gücünü hesaplayınız. *Cevap: 4.9 W*



8-41 Air enters a rectangular duct. The fan power needed to overcome the pressure losses is to be determined. √

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** Air is an ideal gas. **4** The duct involves no components such as bends, valves, and connectors. **5** The flow section involves no work devices such as fans or turbines

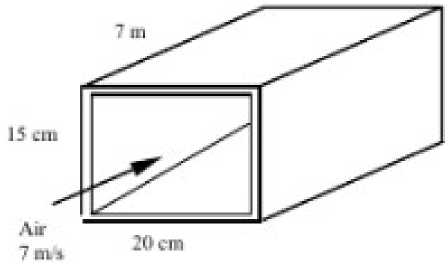
Properties The properties of air at 1 atm and 35°C are ρ = 1.145 kg/m³, μ = 1.895×10⁻⁵ kg/m·s, and ν = 1.655×10⁻⁵ m²/s. The roughness of commercial steel surfaces is ε = 0.000045 m.

Analysis The hydraulic diameter, the volume flow rate, and the Reynolds number in this case are

$$D_h = \frac{4A_c}{p} = \frac{4ab}{2(a+b)} = \frac{4(0.15\text{ m})(0.20\text{ m})}{2(0.15+0.20)\text{ m}} = 0.1714\text{ m}$$

$$\dot{V} = VA_c = V(a\times b) = (7\text{ m/s})(0.15\times 0.20\text{ m}^2) = 0.21\text{ m}^3/\text{s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(1.145\text{ kg/m}^3)(7\text{ m/s})(0.1714\text{ m})}{1.895\times 10^{-5}\text{ kg/m}\cdot\text{s}} = 72{,}490$$



which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon / D_h = \frac{4.5\times 10^{-5}\text{ m}}{0.1714\text{ m}} = 2.625\times 10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon / D_h}{3.7} + \frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{2.625\times 10^{-4}}{3.7} + \frac{2.51}{72{,}490\sqrt{f}}\right)$$

It gives f = 0.02034. Then the pressure drop in the duct and the required pumping power become

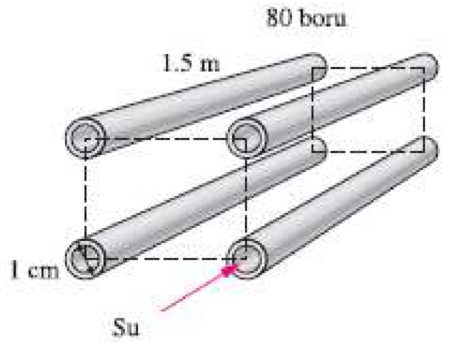
$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.02034\frac{7\text{ m}}{0.1714\text{ m}}\frac{(1.145\text{ kg/m}^3)(7\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ Pa}}{1\text{ N/m}^2}\right) = 23.3\text{ Pa}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.21\text{ m}^3/\text{s})(23.3\text{ Pa})\left(\frac{1\text{ W}}{1\text{ Pa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{4.9\ W}$$

Discussion The friction factor could also be determined easily from the explicit Haaland relation. It would give f = 0.02005, which is sufficiently close to 0.02034. Also, the power input determined is the mechanical power that needs to be imparted to the fluid. The shaft power will be much more than this due to fan inefficiency; the electrical power input will be even more due to motor inefficiency.

8–50 İki akışkan arasında ısı geçişini sağlamada bir gövde içerisine yerleştirilen yüzlerce borudan oluşan gövde-boru tipi ısı değiştiricileri uygulamada yaygın olarak kullanılır. Aktif güneş ışığı ile sıcak su temini sistemindeki böyle bir ısı değiştiricisi, gövdede ve güneş kollektöründe dolaşan su-antifriz çözeltisinden ısıyı alır ve borularda ortalama sıcaklığı 60°C olan ve 15 L/s debi ile akan taze suya verir. Isı değiştiricisinde her birinin iç çapı 1 cm ve uzunluğu 1.5 m olan 80 adet pirinç boru vardır. Giriş, çıkış etkilerini ve yük kayıplarını göz ardı ederek tek bir borudaki basınç düşüşünü ve ısı değiştiricisinin boru tarafındaki akışkan için gerekli pompalama gücünü hesaplayınız.

Uzun süre çalıştıktan sonra, iç yüzeyde eşdeğer pürüzlülüğü 0.4 mm olan 1 mm kalınlığında bir tabaka oluşmaktadır. Aynı pompalama gücü girişi için borularda akan suyun debisindeki yüzde azalmayı hesaplayınız.

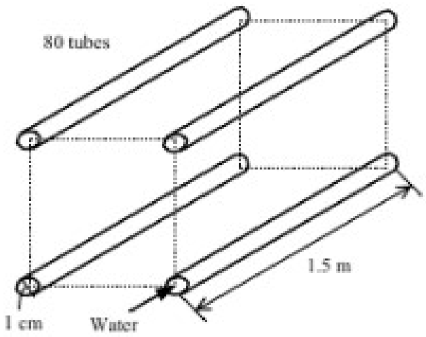


8-50 Water is flowing through a brass tube bank of a heat exchanger at a specified flow rate. The pressure drop and the pumping power required are to be determined. Also, the percent reduction in the flow rate of water through the tubes is to be determined after scale build-up on the inner surfaces of the tubes. √

Assumptions **1** The flow is steady, horizontal, and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed (this is a questionable assumption since the tubes are short, and it will be verified). **3** The inlet, exit, and header losses are negligible, and the tubes involve no components such as bends, valves, and connectors. **4** The piping section involves no work devices such as pumps and turbines.

Properties The density and dynamic viscosity of water at 20°C are ρ = 983.3 kg/m³ and $\mu = 0.467\times10^{-3}$ kg/m·s, respectively. The roughness of brass tubing is 1.5×10^{-6} m.

Analysis First we calculate the average velocity and the Reynolds number to determine the flow regime:



$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{N_{\text{tube}}(\pi D^2/4)} = \frac{0.015\ \text{m}^3/\text{s}}{80[\pi(0.01\ \text{m})^2/4]} = 2.387\ \text{m/s}$$

$$\text{Re} = \frac{\rho V D_h}{\mu} = \frac{(983.3\ \text{kg/m}^3)(2.387\ \text{m/s})(0.01\ \text{m})}{0.467\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 50{,}270$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{1.5\times10^{-6}\ \text{m}}{0.01\ \text{m}} = 1.5\times10^{-4}$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}} = -2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right) \rightarrow \frac{1}{\sqrt{f}} = -2.0\log\left(\frac{1.5\times10^{-4}}{3.7}+\frac{2.51}{50{,}270\sqrt{f}}\right)$$

It gives f = 0.0214. Then the pressure drop, the head loss, and the useful pumping power required become

$$\Delta P = \Delta P_L = f\frac{L}{D}\frac{\rho V^2}{2} = 0.0214\frac{1.5\ \text{m}}{0.01\ \text{m}}\frac{(983.3\ \text{kg/m}^3)(2.387\ \text{m/s})^2}{2}\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1\ \text{kN/m}^2}\right) = \mathbf{8.99\ kPa}$$

$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.015\ \text{m}^3/\text{s})(8.99\ \text{kPa})\left(\frac{1\ \text{kW}}{1\ \text{kPa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{0.135\ kW}$$

Therefore, useful power input in the amount of 0.135 kW is needed to overcome the frictional losses in the tube. The hydrodynamic entry length in this case is

$$L_{\text{h,turbulent}} \approx 10D = 10(0.01\ \text{m}) = 0.1\ \text{m}$$

which is much less than 1.5 m. Therefore, the assumption of fully developed flow is valid. (The effect of the entry region is to increase the friction factor, and thus the pressure drop and pumping power).

8–58 İçi su dolu 3 m yüksekliğinde bir haznenin alt yüzeyine 1.5 cm çapında bir delik delinerek su tahliye edilmek isteniyor. Kinetik enerji düzeltme faktörünün etkisini göz ardı ederek (*a*) delik girişi iyi yuvarlatılmış ise ve (*b*) giriş keskin kenarlı ise delikten akan suyun debisini hesaplayınız.

8-58 Water is to be withdrawn from a water reservoir by drilling a hole at the bottom surface. The flow rate of water through the hole is to be determined for the well-rounded and sharp-edged entrance cases.

Assumptions **1** The flow is steady and incompressible. **2** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **3** The effect of the kinetic energy correction factor is disregarded, and thus $\alpha = 1$.

Analysis The loss coefficient is $K_L = 0.5$ for the sharp-edged entrance, and $K_L = 0.03$ for the well-rounded entrance.

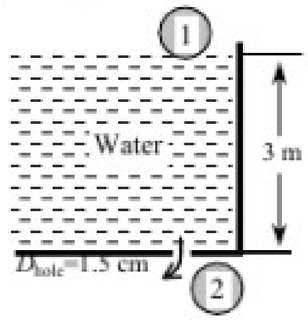
We take point 1 at the free surface of the reservoir and point 2 at the exit of the hole. We also take the reference level at the exit of the hole ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is zero ($V_1 = 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump,u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine,e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where the head loss is expressed as $h_L = K_L\dfrac{V^2}{2g}$. Substituting and solving for V_2 gives

$$z_1=\alpha_2\frac{V_2^2}{2g}+K_L\frac{V_2^2}{2g} \quad\rightarrow\quad 2gz_1=V_2^2(\alpha_2+K_L) \quad\rightarrow\quad V_2=\sqrt{\frac{2gz_1}{\alpha_2+K_L}}=\sqrt{\frac{2gz_1}{1+K_L}}$$

since $\alpha_2 = 1$. Note that in the special case of $K_L = 0$, it reduces to the Toricelli equation $V_2=\sqrt{2gz_1}$, as expected. Then the volume flow rate becomes



$$\dot{V}=A_cV_2=\frac{\pi D_{hole}^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}$$

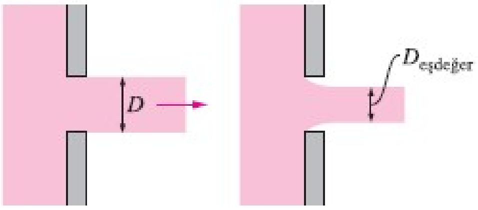
Substituting the numerical values, the flow rate for both cases are determined to be

Well-rounded entrance: $$\dot{V}=\frac{\pi D_{hole}^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}=\frac{\pi(0.015\text{ m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(3\text{ m})}{1+0.03}}=\mathbf{1.34\times10^{-3}\ m^3/s}$$

Sharp-edged entrance: $$\dot{V}=\frac{\pi D_{hole}^2}{4}\sqrt{\frac{2gz_1}{1+K_L}}=\frac{\pi(0.015\text{m})^2}{4}\sqrt{\frac{2(9.81\text{ m/s}^2)(3\text{ m})}{1+0.5}}=\mathbf{1.11\times10^{-3}\ m^3/s}$$

Discussion The flow rate in the case of frictionless flow ($K_L = 0$) is 1.36×10^{-3} m^3/s. Note that the frictional losses cause the flow rate to decrease by 1.5% for well-rounded entrance, and 18.5% for the sharp-edged entrance.

8–59 Bir su deposunun serbest yüzeyinden H kadar aşağıda, yan duvardaki D çaplı dairesel delikten akan suyu ele alınız. Keskin kenarlı girişi olan gerçek bir delikteki (K_K) debi, "sürtünmesiz" akış ve dolayısıyla delikteki kayıp ihmal edilerek hesaplanan debiden önemli oranda daha az olacaktır. Kinetik enerji düzeltme faktörünün etkisini göz ardı ederek, sürtünmesiz akış bağıntılarında kullanmak üzere, keskin kenarlı deliğin "eşdeğer çapı" için bir bağıntı elde ediniz.



8-59 Water is discharged from a water reservoir through a circular hole of diameter D at the side wall at a vertical distance H from the free surface. A relation for the "equivalent diameter" of the sharp-edged hole for use in frictionless flow relations is to be obtained.

Assumptions **1** The flow is steady and incompressible. **2** The reservoir is open to the atmosphere so that the pressure is atmospheric pressure at the free surface. **3** The effect of the kinetic energy correction factor is disregarded, and thus $\alpha = 1$.

Analysis The loss coefficient is $K_L = 0.5$ for the sharp-edged entrance, and $K_L = 0$ for the "frictionless" flow. We take point 1 at the free surface of the reservoir and point 2 at the exit of the hole, which is also taken to be the reference level ($z_2 = 0$). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is zero ($V_1 = 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{pump,u}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{turbin\,e,e}+h_L \quad\to\quad H=\alpha_2\frac{V_2^2}{2g}+h_L$$

where the head loss is expressed as $h_L = K_L\dfrac{V^2}{2g}$. Substituting and solving for V_2 gives

$$H=\alpha_2\frac{V_2^2}{2g}+K_L\frac{V_2^2}{2g} \quad\to\quad 2gH=V_2^2(\alpha_2+K_L) \quad\to\quad V_2=\sqrt{\frac{2gH}{\alpha_2+K_L}}=\sqrt{\frac{2gH}{1+K_L}}$$

since $\alpha_2 = 1$. Then the volume flow rate becomes

$$\dot{V}=A_cV_2=\frac{\pi D^2}{4}\sqrt{\frac{2gH}{1+K_L}} \qquad (1)$$

Note that in the special case of $K_L = 0$ (frictionless flow), the velocity relation reduces to the Toricelli equation, $V_{2,frictionless}=\sqrt{2gH}$. The flow rate in this case through a hole of D_e (equivalent diameter) is

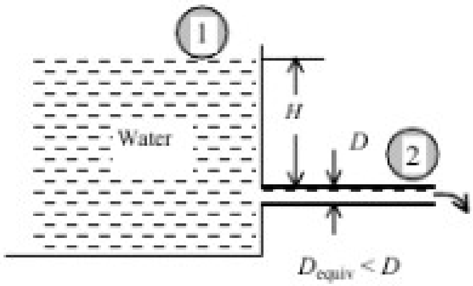
$$\dot{V}=A_{c,equiv}V_{2,frictionless}=\frac{\pi D_{equiv}^2}{4}\sqrt{2gH} \qquad (2)$$

Setting Eqs. (1) and (2) equal to each other gives the desired relation for the equivalent diameter,

$$\frac{\pi D_{equiv}^2}{4}\sqrt{2gH}=\frac{\pi D^2}{4}\sqrt{\frac{2gH}{1+K_L}}$$

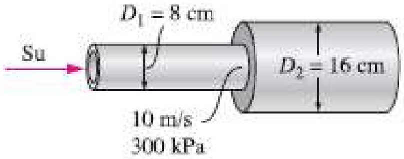
which gives

$$D_{equiv}=\frac{D}{(1+K_L)^{1/4}}=\frac{D}{(1+0.5)^{1/4}}=0.904D$$



Discussion Note that the effect of frictional losses of a sharp-edged entrance is to reduce the diameter by about 10%. Also, noting that the flow rate is proportional to the square of the diameter, we have $\dot{V} \propto D_{equiv}^2 = (0.904D)^2 = 0.82D^2$. Therefore, the flow rate through a sharp-edged entrance is about 18% less compared to the frictionless entrance case.

8–61 Yatay bir boru D_1 = 8 cm'den D_2 = 16 cm'ye ani olarak genişlemektedir. Küçük borudaki su hızı 10 m/s' dir ve akış türbülanslı olup basınç P_1 = 300 kPa'dır. Girişte ve çıkışta kinetik enerji düzeltme faktörünü 1.06 alarak aşağıakım basıncı P_2'yi hesaplayınız. Eğer Bernoulli denklemi kullanılmış olsaydı yapılan hata ne olurdu. *Cevaplar:* 322 kPa, 25 kPa



8-61 A horizontal water pipe has an abrupt expansion. The water velocity and pressure in the smaller diameter pipe are given. The pressure after the expansion and the error that would have occurred if the Bernoulli Equation had been used are to be determined.

Assumptions **1**The flow is steady, horizontal, and incompressible. **2** The flow at both the inlet and the outlet is fully developed and turbulent with kinetic energy corrections factors of $\alpha_1 = \alpha_2 = 1.06$ (given).

Properties We take the density of water to be ρ = 1000 kg/m^3.

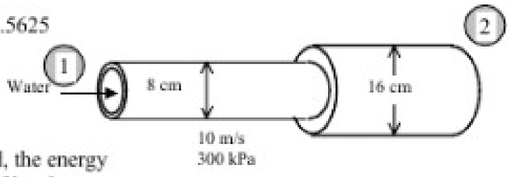
Analysis Noting that ρ = const. (incompressible flow), the downstream velocity of water is

$$\dot{m}_1 = \dot{m}_2 \rightarrow \rho V_1 A_1 = \rho V_2 A_2 \rightarrow V_2 = \frac{A_1}{A_2}V_1 = \frac{\pi D_1^2/4}{\pi D_2^2/4}V_1 = \frac{D_1^2}{D_2^2}V_1 = \frac{(0.08\text{ m})^2}{(0.16\text{ m})^2}(10\text{ m/s}) = 2.5\text{ m/s}$$

The loss coefficient for sudden expansion and the head loss can be calculated from

$$K_L = \left(1-\frac{A_{\text{small}}}{A_{\text{large}}}\right)^2 = \left(1-\frac{D_1^2}{D_2^2}\right)^2 = \left(1-\frac{0.08^2}{0.16^2}\right)^2 = 0.5625$$

$$h_L = K_L\frac{V_1^2}{2g} = (0.5625)\frac{(10\text{ m/s})^2}{2(9.81\text{ m/s}^2)} = 2.87\text{ m}$$



Noting that $z_1 = z_2$ and there are no pumps or turbines involved, the energy equation for the expansion section can be expressed in terms of heads as

$$\frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} + z_1 + h_{\text{pump,u}} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + z_2 + h_{\text{turbine,e}} + h_L \quad \rightarrow \quad \frac{P_1}{\rho g} + \alpha_1\frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \alpha_2\frac{V_2^2}{2g} + h_L$$

Solving for P_2 and substituting,

$$P_2 = P_1 + \rho\left\{\frac{\alpha_1 V_1^2 - \alpha_2 V_2^2}{2} - gh_L\right\}$$

$$= (300\text{ kPa}) + (1000\text{ kg/m}^3)\left\{\frac{1.06(10\text{ m/s})^2 - 1.06(2.5\text{ m/s})^2}{2} - (9.81\text{ m/s}^2)(2.87\text{ m})\right\}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)$$

$$= \mathbf{322\ kPa}$$

Therefore, despite the head (and pressure) loss, the pressure increases from 300 kPa to 321 kPa after the expansion. This is due to the conversion of dynamic pressure to static pressure when the velocity is decreased.

When the head loss is disregarded, the downstream pressure is determined from the Bernoulli equation to be

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad \frac{P_1}{\rho g} + \frac{V_1^2}{2g} = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} \quad \rightarrow \quad P_1 = P_1 + \rho\frac{V_1^2 - V_2^2}{2}$$

Substituting,

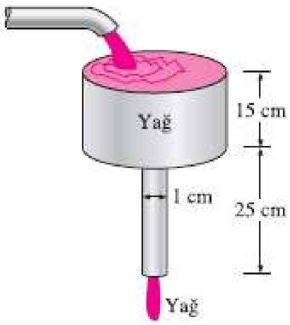
$$P_2 = (300\text{ kPa}) + (1000\text{ kg/m}^3)\frac{(10\text{ m/s})^2 - (2.5\text{ m/s})^2}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right) = 347\text{ kPa}$$

Therefore, the error in the Bernoulli equation is $\qquad$ Error = $P_{2,\text{ Bernoulli}} - P_2$ = 347 – 322 = **25 kPa**

Note that the use of the Bernoulli equation results in an error of (347 – 322)/322 = 0.078 or 7.8%.

Discussion It is common knowledge that higher pressure upstream is necessary to cause flow, and it may come as a surprise that the downstream pressure has *increased* after the abrupt expansion, despite the loss. This is because the sum of the three Bernoulli terms which comprise the total head, consisting of pressure head, velocity head, and elevation head, namely $[P/\rho g + ½V^2/g + z]$, drives the flow. With a geometric flow expansion, initially higher velocity head is converted to downstream pressure head, and this increase outweighs the non-convertible and non-recoverable head loss term.

8–73 Sıcaklığı 20°C olan yağ, düşey bir cam huniden akmaktadır. Huni, 15 cm yüksekliğinde silindirik bir depo ve 1 cm çapında 25 cm yüksekliğinde bir borudan oluşmaktadır. Huniye bir tanktan yağ ilave edilerek devamlı dolu kalması sağlanmaktadır. Giriş etkilerini ihmal ederek hunideki yağ debisini belirleyiniz ayrıca hunideki gerçek debinin "sürtünmesiz" durumdaki maksimum debiye oranı olarak tanımlanan "huni etkinliğini" hesaplayınız. *Cevaplar:* 4.09×10^{-6} m³/s, yüzde 1.86



8-73 Oil is flowing through a vertical glass funnel which is always maintained full. The flow rate of oil through the funnel and the funnel effectiveness are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed (to be verified). **3** The frictional loses in the cylindrical reservoir are negligible since its diameter is very large and thus the oil velocity is very low.

Properties The density and viscosity of oil at 20°C are ρ= 888.1 kg/m³ and μ = 0.8374 kg/m·s.

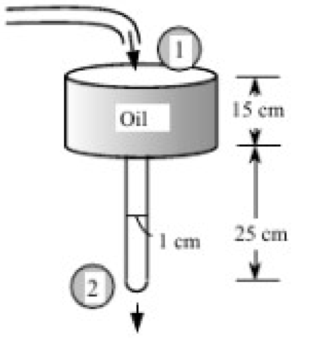
Analysis We take point 1 at the free surface of the oil in the cylindrical reservoir, and point 2 at the exit of the funnel pipe which is also taken as the reference level ($z_2 = 0$). The fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is negligible ($V_1 \cong 0$). For the ideal case of "frictionless flow," the exit velocity is determined from the Bernoulli equation to be

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad \rightarrow \quad V_2 = V_{2,\max} = \sqrt{2gz_1}$$

Substituting,

$$V_{2,\max} = \sqrt{2gz_1} = \sqrt{2(9.81\ \text{m/s}^2)(0.40\ \text{m})} = 2.801\ \text{m/s}$$

This is the flow velocity for the frictionless case, and thus it is the maximum flow velocity. Then the maximum flow rate and the Reynolds number become



$$\dot{V}_{\max} = V_{2,\max}A_2 = V_{2,\max}(\pi D_2^2/4)$$
$$= (2.801\ \text{m/s})[\pi(0.01\ \text{m})^2/4] = 2.20\times10^{-4}\ \text{m}^3/\text{s}$$

$$\text{Re} = \frac{\rho VD}{\mu} = \frac{(888.1\ \text{kg/m}^3)(2.801\ \text{m/s})(0.01\ \text{m})}{0.8374\ \text{kg/m}\cdot\text{s}} = 29.71$$

which is less than 2300. Therefore, the flow is laminar, as postulated. (Note that in the actual case the velocity and thus the Reynolds number will be even smaller, verifying the flow is always laminar). The entry length in this case is

$$L_h = 0.05\,\text{Re}\,D = 0.05\times29.71\times(0.01\ \text{m}) = 0.015\ \text{m}$$

which is much less than the 0.25 m pipe length. Therefore, the entrance effects can be neglected as postulated.

Noting that the flow through the pipe is laminar and can be assumed to be fully developed, the flow rate can be determined from the appropriate relation with θ = -90° since the flow is downwards in the vertical direction,

$$\dot{V} = \frac{(\Delta P - \rho g L\sin\theta)\pi D^4}{128\mu L}$$

where $\Delta P = P_{\text{pipe inlet}} - P_{\text{pipe exit}} = (P_{atm} + \rho g h_{\text{cylinder}}) - P_{atm} = \rho g h_{\text{cylinder}}$ is the pressure difference across the pipe, $L = h_{\text{pipe}}$, and $\sin\theta = \sin(-90°) = -1$. Substituting, the flow rate is determined to be

$$\dot{V} = \frac{\rho g(h_{\text{cylinder}} + h_{\text{pipe}})\pi D^4}{128\mu L} = \frac{(888.1\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.15+0.25\ \text{m})\pi(0.01\ \text{m})^4}{128(0.8374\ \text{kg/m}\cdot\text{s})(0.25\ \text{m})} = \mathbf{4.09\times10^{-6}\ m^3/s}$$

Then the "funnel effectiveness" becomes

$$\text{Eff} = \frac{\dot{V}}{\dot{V}_{\max}} = \frac{4.09\times10^{-6}\ \text{m}^3/\text{s}}{2.20\times10^{-4}\ \text{m}^3/\text{s}} = 0.0186 \quad \text{or} \quad \mathbf{1.86\%}$$

Discussion Note that the flow is driven by gravity alone, and the actual flow rate is a small fraction of the flow rate that would have occurred if the flow were frictionless.

8–78 Bir tank, güneşle ısıtılan 40°C'deki su ile doldurulmuştur. Bu su, yerçekimi etkisiyle akış oluşturarak bir tarladaki duşlar için kullanılacaktır. Sistem 20 m uzunluğunda 1.5 cm çapında galvanizli demir boru ve vanasız 4 köşe dönüş (90°) ve tam açık küresel vanadan oluşmaktadır. Suyun 0.7 L/s'lik debi ile duş başlığından akması istendiğine göre, tanktaki su seviyesinin duştan çıkış seviyesinden ne kadar yüksekte olması gerektiğini hesaplayınız. Giriş ve duş başlığı kayıplarını ve kinetik enerji düzeltme faktörünü etkisini ihmal ediniz.

8-78 A solar heated water tank is to be used for showers using gravity driven flow. For a specified flow rate, the elevation of the water level in the tank relative to showerhead is to be determined.

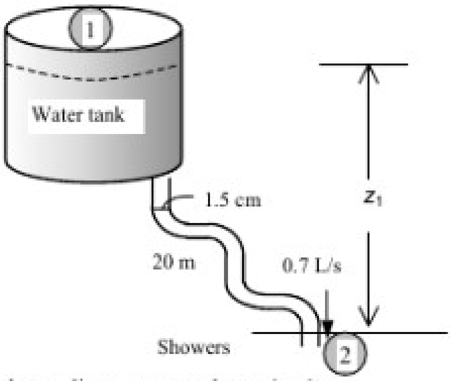
Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** The elevation difference between the free surface of water in the tank and the shower head remains constant. **5** There are no pumps or turbines in the piping system. **6** The losses at the entrance and at the showerhead are said to be negligible. **7** The water tank is open to the atmosphere. **8** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Properties The density and dynamic viscosity of water at 40°C are $\rho = 992.1$ kg/m^3 and $\mu = 0.653\times10^{-3}$ kg/m·s, respectively. The loss coefficient is $K_L = 0.5$ for a sharp-edged entrance. The roughness of galvanized iron pipe is $\varepsilon = 0.00015$ m.

Analysis The piping system involves 20 m of 1.5-cm diameter piping, an entrance with negligible loss, 4 miter bends (90°) without vanes ($K_L = 1.1$ each), and a wide open globe valve ($K_L = 10$). We choose point 1 at the free surface of water in the tank, and point 2 at the shower exit, which is also taken to be the reference level ($z_2 = 0$). The fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$), and $V_1 = 0$. Then the energy equation for a control volume between these two points simplifies to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump, u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine, e}}+h_L \quad\rightarrow\quad z_1=\alpha_2\frac{V_2^2}{2g}+h_L$$

where $h_L = h_{L,\text{total}} = h_{L,\text{major}} + h_{L,\text{minor}} = \left(f\dfrac{L}{D}+\sum K_L\right)\dfrac{V_2^2}{2g}$



since the diameter of the piping system is constant. The average velocity in the pipe and the Reynolds number are

$$V_2=\frac{\dot V}{A_c}=\frac{\dot V}{\pi D^2/4}=\frac{0.0007\ \text{m}^3/\text{s}}{\pi(0.015\ \text{m})^2/4}=3.961\ \text{m/s}$$

$$\text{Re}=\frac{\rho V_2 D}{\mu}=\frac{(992.1\ \text{kg/m}^3)(3.961\ \text{m/s})(0.015\ \text{m})}{0.653\times10^{-3}\ \text{kg/m}\cdot\text{s}}=90{,}270$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00015\ \text{m}}{0.015\ \text{m}}=0.01$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt f}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt f}\right)\ \rightarrow\ \frac{1}{\sqrt f}=-2.0\log\left(\frac{0.0051}{3.7}+\frac{2.51}{90{,}270\sqrt f}\right)$$

It gives $f = 0.03857$. The sum of the loss coefficients is

$$\sum K_L = K_{L,\text{entrance}}+4K_{L,\text{elbow}}+K_{L,\text{valve}}+K_{L,\text{exit}}=0+4\times1.1+10+0=14.4$$

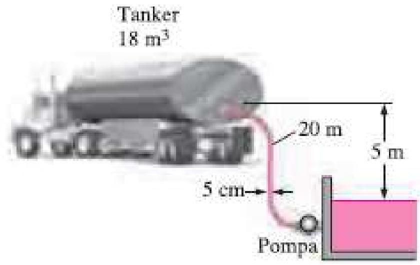
Note that we do not consider the exit loss unless the exit velocity is dissipated within the system considered (in this case it is not). Then the total head loss and the elevation of the source become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.03857)\frac{20\ \text{m}}{0.015\ \text{m}}+14.4\right)\frac{(3.961\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)}=52.6\ \text{m}$$

$$z_1=\alpha_2\frac{V_2^2}{2g}+h_L=(1)\frac{(3.961\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)}+52.6\ \text{m}=\mathbf{53.4\ m}$$

since $\alpha_2 = 1$. Therefore, the free surface of the tank must be 53.4 m above the shower exit to ensure water flow at the specified rate.

çıkışında kinetik enerji düzeltme faktörünü 1.05 alarak ve toplam pompa verimini yüzde 82 kabul ederek pompaya verilmesi gereken gücü hesaplayınız.



8–80 Bir tanker, yoğunluğu $\rho = 920$ kg/m^3 ve viskozitesi $\mu = 0.045$ kg/m·s olan akaryakıt ile doldurulacaktır. Dolum işlemi yeraltı deposundan 20 m uzunluğunda 5 cm çapında hafif yuvarlatılmış girişi ve 90°'lik iki tane pürüzsüz dönüşü olan plastik hortum kullanarak yapılmaktadır. Depodaki yağ seviyesi ile hortumun bağlandığı tankerdeki yağ seviyesi arasındaki yükseklik farkı 5 m'dir. Tanker

8-80 A tanker is to be filled with fuel oil from an underground reservoir using a plastic hose. The required power input to the pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** Fuel oil level remains constant. **5** Reservoir is open to the atmosphere.

Properties The density and dynamic viscosity of fuel oil are given to be ρ = 920 kg/m^3 and μ = 0.045 kg/m-s. The loss coefficient is K_L = 0.12 for a slightly-rounded entrance and K_L = 0.3 for a 90° smooth bend (flanged). The plastic pipe is smooth and thus ε = 0. The kinetic energy correction factor at hose discharge is given to be α = 1.05.

Analysis We choose point 1 at the free surface of oil in the reservoir and point 2 at the exit of the hose in the tanker. We note the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and the fluid velocity at point 1 is zero ($V_1 = 0$). We take the free surface of the reservoir as the reference level ($z_1 = 0$). Then the energy equation for a control volume between these two points simplifies to

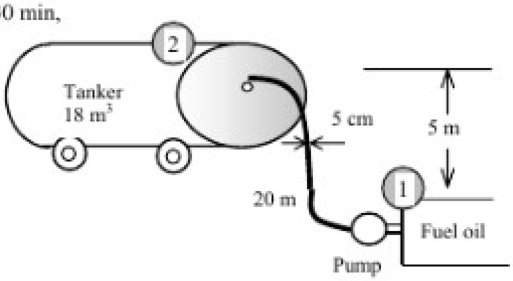
$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump, u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine, e}}+h_L \quad\rightarrow\quad h_{\text{pump, u}}=\alpha_2\frac{V_2^2}{2g}+z_2+h_L$$

where

$$h_L=h_{L,\text{total}}=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$

since the diameter of the piping system is constant. The flow rate is determined from the requirement that the tanker must be filled in 30 min,

$$\dot{V}=\frac{V_{\text{tanker}}}{\Delta t}=\frac{18\text{ m}^3}{(30\times 60\text{ s})}=0.01\text{ m}^3/\text{s}$$



Then the average velocity in the pipe and the Reynolds number become

$$V_2=\frac{\dot{V}}{A_c}=\frac{\dot{V}}{\pi D^2/4}=\frac{0.01\text{ m}^3/\text{s}}{\pi(0.05\text{ m})^2/4}=5.093\text{ m/s}$$

$$\text{Re}=\frac{\rho V_2 D}{\mu}=\frac{(920\text{ kg/m}^3)(5.093\text{ m/s})(0.05\text{ m})}{0.045\text{ kg/m}\cdot\text{s}}=5206$$

which is greater than 4000. Therefore, the flow is turbulent. The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation,

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\quad\rightarrow\quad\frac{1}{\sqrt{f}}=-2.0\log\left(0+\frac{2.51}{5206\sqrt{f}}\right)$$

It gives f = 0.0370. The sum of the loss coefficients is

$$\sum K_L=K_{L,\text{entrance}}+2K_{L,\text{bend}}=0.12+2\times 0.3=0.72$$

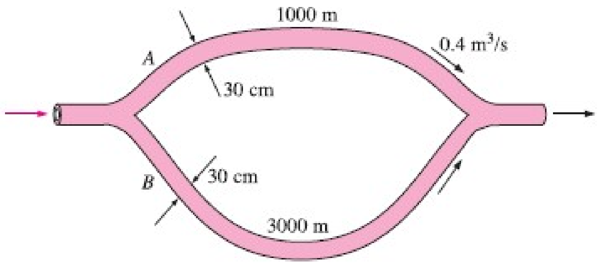
Note that we do not consider the exit loss unless the exit velocity is dissipated within the system (in this case it is not). Then the total head loss, the useful pump head, and the required pumping power become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.0370)\frac{20\text{ m}}{0.05\text{ m}}+0.72\right)\frac{(5.093\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=20.5\text{ m}$$

$$h_{\text{pump, u}}=\frac{V_2^2}{2g}+z_2+h_L=1.05\frac{(5.093\text{ m/s})^2}{2(9.81\text{ m/s}^2)}+5\text{ m}+20.5\text{ m}=26.9\text{ m}$$

$$\dot{W}_{\text{pump}}=\frac{\dot{V}\rho g h_{\text{pump, u}}}{\eta_{\text{pump}}}=\frac{(0.01\text{ m}^3/\text{s})(920\text{ kg/m}^3)(9.81\text{ m/s}^2)(26.9\text{ m})}{0.82}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1\text{ kN}\cdot\text{m/s}}\right)=\mathbf{2.96\text{ kW}}$$

Discussion Note that the minor losses in this case are negligible (0.72/15.52 = 0.046 or about 5% of total losses). Also, the friction factor could be determined easily from the Haaland relation (it gives 0.0372).



ŞEKİL P8–82

8–82 Dökme demirden yapılan su dağıtım sisteminin belirli bir bölümünde paralel bir kısım vardır. Birbirine paralel iki borunun çapı 30 cm'dir ve akış tamamen türbülanslıdır. Dallardan birisi (*A* borusu) 1000 m uzunluğunda diğeri ise (*B* borusu) 3000 m uzunluğundadır. Eğer *A* borusundaki debi 0.4 m^3/s ise *B* borusundaki debiyi bulunuz. Yerel kayıpları göz ardı ediniz ve su sıcaklığını 15°C kabul alınız. Akışın tamamen türbülanslı olduğunu ve dolayısıyla sürtünme faktörünün Reynolds sayısından bağımsız olduğunu gösteriniz. *Cevap:* 0.231 m^3/s

8–83 *A* borusunda yarı kapalı sürgülü vana ($K_K = 2.1$) ve *B* borusunda tam açık küresel vana ($K_K = 10$) bulunduğunu ve yerel kayıpların ihmal edilebilir olduğunu kabul ederek Problem 8–82'yi tekrar çözünüz. Akışı tamamen türbülanslı kabul ediniz.

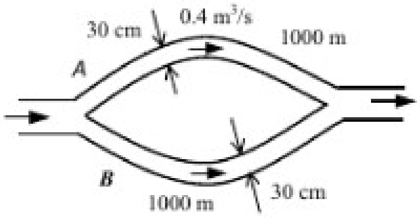
8-82 Cast iron piping of a water distribution system involves a parallel section with identical diameters but different lengths. The flow rate through one of the pipes is given, and the flow rate through the other pipe is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The entrance effects are negligible, and thus the flow is fully developed. **3** The minor losses are negligible. **4** The flow is fully turbulent and thus the friction factor is independent of the Reynolds number (to be verified).

Properties The density and dynamic viscosity of water at 15°C are ρ = 999.1 kg/m^3 and μ =1.138×10^{-3} kg/m·s. The roughness of cast iron pipe is ε = 0.00026 m.

Analysis The average velocity in pipe *A* is

$$V_A = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.4\ \text{m}^3/\text{s}}{\pi(0.30\ \text{m})^2/4} = 5.659\ \text{m/s}$$



When two pipes are parallel in a piping system, the head loss for each pipe must be same. When the minor losses are disregarded, the head loss for fully developed flow in a pipe of length *L* and diameter *D* is

$$h_L = f\frac{L}{D}\frac{V^2}{2g}$$

Writing this for both pipes and setting them equal to each other, and noting that $D_A = D_B$ (given) and $f_A = f_B$ (to be verified) gives

$$f_A\frac{L_A}{D_A}\frac{V_A^2}{2g} = f_B\frac{L_B}{D_B}\frac{V_B^2}{2g} \quad\rightarrow\quad V_B = V_A\sqrt{\frac{L_A}{L_B}} = (5.659\ \text{m/s})\sqrt{\frac{1000\ \text{m}}{3000\ \text{m}}} = 3.267\ \text{m/s}$$

Then the flow rate in pipe *B* becomes

$$\dot{V}_B = A_c V_B = [\pi D^2/4]V_B = [\pi(0.30\ \text{m})^2/4](3.267\ \text{m/s}) = \mathbf{0.231\ m^3/s}$$

***Proof** that flow is fully turbulent and thus friction factor is independent of Reynolds number:*

The velocity in pipe *B* is lower. Therefore, if the flow is fully turbulent in pipe *B*, then it is also fully turbulent in pipe *A*. The Reynolds number in pipe *B* is

$$\text{Re}_B = \frac{\rho V_B D}{\mu} = \frac{(999.1\ \text{kg/m}^3)(3.267\ \text{m/s})(0.30\ \text{m})}{1.138\times10^{-3}\ \text{kg/m}\cdot\text{s}} = 0.860\times10^6$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D = \frac{0.00026\ \text{m}}{0.30\ \text{m}} = 0.00087$$

From Moody's chart, we observe that for a relative roughness of 0.00087, the flow is fully turbulent for Reynolds number greater than about 10^6. Therefore, the flow in both pipes is fully turbulent, and thus the assumption that the friction factor is the same for both pipes is valid.

Discussion Note that the flow rate in pipe *B* is less than the flow rate in pipe *A* because of the larger losses due to the larger length.

8–87 Büyük binalarda su tankındaki sıcak su bir kapalı devrede dolaştırılır ve böylece bir kişi sıcak su gelmeden önce uzun borulardaki bütün suyun akmasını beklemeye mecbur kalmaz. Dolaşım devresinde 40 m uzunluğunda 1.2 cm çapında dökme demirden boru ve altı tane 90° dişli pürüzsüz dönüş ve iki tane tam açık sürgülü vana vardır. Devredeki ortalama akış hızı 2.5 m/s olduğuna göre sirkülasyon pompası için gereken gücü hesaplayınız. Ortalama su sıcaklığını 60°C ve pompa verimini yüzde 70 alınız. *Cevap:* 0.217 kW

8-87 Hot water in a water tank is circulated through a loop made of cast iron pipes at a specified average velocity. The required power input for the recirculating pump is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The flow is fully developed. **3** The flow is turbulent so that the tabulated value of the loss coefficients can be used (to be verified). **4** Minor losses other than those for elbows and valves are negligible.

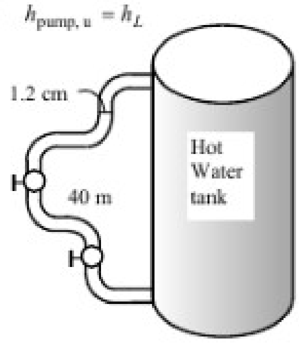
Properties The density and dynamic viscosity of water at 60°C are $\rho = 983.3$ kg/m^3, $\mu = 0.467\times10^{-3}$ kg/m·s. The roughness of cast iron pipes is 0.00026 m. The loss coefficient is $K_L = 0.9$ for a threaded 90° smooth bend and $K_L = 0.2$ for a fully open gate valve.

Analysis Since the water circulates continually and undergoes a cycle, we can take the entire recirculating system as the control volume, and choose points 1 and 2 at any location at the same point. Then the properties (pressure, elevation, and velocity) at 1 and 2 will be identical, and the energy equation will simplify to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump, u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine, e}}+h_L \quad\rightarrow\quad h_{\text{pump, u}}=h_L$$

where

$$h_L=h_{L,\text{major}}+h_{L,\text{minor}}=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}$$



since the diameter of the piping system is constant. Therefore, the pumping power is to be used to overcome the head losses in the flow. The flow rate and the Reynolds number are

$$\dot{V}=VA_c=V(\pi D^2/4)=(2.5\text{ m/s})[\pi(0.012\text{ m})^2/4]=2.83\times10^{-4}\text{ m}^3/\text{s}$$

$$\text{Re}=\frac{\rho VD}{\mu}=\frac{(983.3\text{ kg/m}^3)(2.5\text{ m/s})(0.012\text{ m})}{0.467\times10^{-3}\text{ kg/m}\cdot\text{s}}=63{,}200$$

which is greater than 4000. Therefore, the flow is turbulent. The relative roughness of the pipe is

$$\varepsilon/D=\frac{0.00026\text{ m}}{0.012\text{ m}}=0.0217$$

The friction factor can be determined from the Moody chart, but to avoid the reading error, we determine it from the Colebrook equation using an equation solver (or an iterative scheme),

$$\frac{1}{\sqrt{f}}=-2.0\log\left(\frac{\varepsilon/D}{3.7}+\frac{2.51}{\text{Re}\sqrt{f}}\right)\ \rightarrow\ \frac{1}{\sqrt{f}}=-2.0\log\left(\frac{0.0217}{3.7}+\frac{2.51}{63200\sqrt{f}}\right)$$

It gives $f = 0.05075$. Then the total head loss, pressure drop, and the required pumping power input become

$$h_L=\left(f\frac{L}{D}+\sum K_L\right)\frac{V_2^2}{2g}=\left((0.05075)\frac{40\text{ m}}{0.012\text{ m}}+6\times0.9+2\times0.2\right)\frac{(2.5\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=55.8\text{ m}$$

$$\Delta P=\Delta P_L=\rho g h_L=(983.3\text{ kg/m}^3)(9.81\text{ m/s}^2)(55.8\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}}\right)\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)=538\text{ kPa}$$

$$\dot{W}_{\text{elect}}=\frac{\dot{W}_{\text{pump, u}}}{\eta_{\text{pump-motor}}}=\frac{\dot{V}\Delta P}{\eta_{\text{pump-motor}}}=\frac{(2.83\times10^{-4}\text{ m}^3/\text{s})(538\text{ kPa})}{0.70}\left(\frac{1\text{ kW}}{1\text{ kPa}\cdot\text{m}^3/\text{s}}\right)=\mathbf{0.217\ kW}$$

Therefore, the required power input of the recirculating pump is 0.217 kW

Discussion It can be shown that the required pumping power input for the recirculating pump is 0.210 kW when the minor losses are not considered. Therefore, the minor losses can be neglected in this case without a major loss in accuracy.

8–98 Sıcaklığı 10°C olan amonyağın (ρ = 624.6 kg/m^3 ve μ = 1.697 $\times$ 10^{-4} kg/m · s) 3 cm çaplı borudaki debisi, diferansiyel basınç ölçer ile tertibatlandırılmış 1.5 cm çaplı akış lülesi ile ölçülmektedir. Diferansiyel basınç ölçerde 50 kPa'lık basınç farkı okunduğuna göre borudaki amonyak debisini ve ortalama akış hızını hesaplayınız.

8-98 The flow rate of ammonia is to be measured with flow nozzle equipped with a differential pressure gage. For a given pressure drop, the flow rate and the average flow velocity are to be determined.

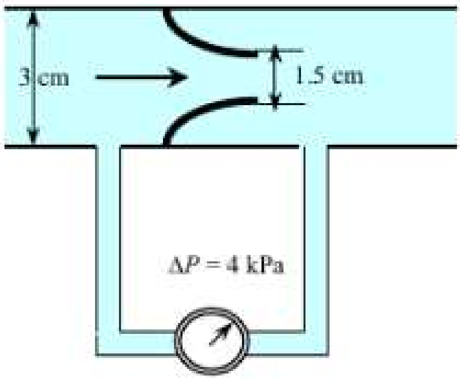
Assumptions **1** The flow is steady and incompressible. **2** The discharge coefficient of the nozzle meter is C_d = 0.96.

Properties The density and dynamic viscosity of ammonia are given to be ρ = 624.6 kg/m^3 and μ = 1.697×10^{-4} kg/m·s, respectively.

Analysis The diameter ratio and the throat area of the meter are

$$\beta = d / D = 1.5 / 3 = 0.50$$

$$A_0 = \pi d^2 / 4 = \pi (0.015\text{ m})^2 / 4 = 1.767\times10^{-4}\text{ m}^2$$



Noting that ΔP = 4 kPa = 4000 N/m^2, the flow rate becomes

$$\dot{V} = A_0 C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

$$= (1.767\times10^{-4}\text{ m}^2)(0.96)\sqrt{\frac{2\times4000\text{ N/m}^2}{(624.6\text{ kg/m}^3)((1-0.50^4)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}$$

$$= \mathbf{0.627\times10^{-3}\ m^3/s}$$

which is equivalent to 0.627 L/s. The average flow velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.627\times10^{-3}\text{ m}^3/\text{s}}{\pi(0.03\text{ m})^2/4} = \mathbf{0.887\ m/s}$$

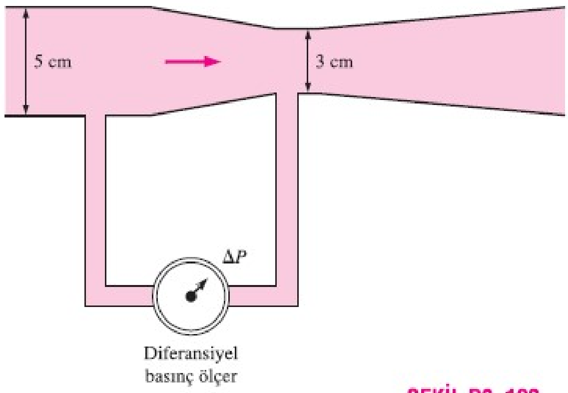
Discussion The Reynolds number of flow through the pipe is

$$\text{Re} = \frac{\rho V D}{\mu} = \frac{(624.6\text{ kg/m}^3)(0.887\text{ m/s})(0.03\text{ m})}{1.697\times10^{-4}\text{ kg/m}\cdot\text{s}} = 9.79\times10^4$$

Substituting the β and Re values into the orifice discharge coefficient relation gives

$$C_d = 0.9975 - \frac{6.53\beta^{0.5}}{\text{Re}^{0.5}} = 0.9975 - \frac{6.53(0.50)^{0.5}}{(9.79\times10^4)^{0.5}} = 0.983$$

which is about 2% different than the assumed value of 0.96. Using this refined value of C_d, the flow rate becomes 0.642 L/s, which differs from our original result by only 2.4%. If the problem is solved using an equation solver such as EES, then the problem can be formulated using the curve-fit formula for C_d (which depends on Reynolds number), and all equations can be solved simultaneously by letting the equation solver perform the iterations as necessary.



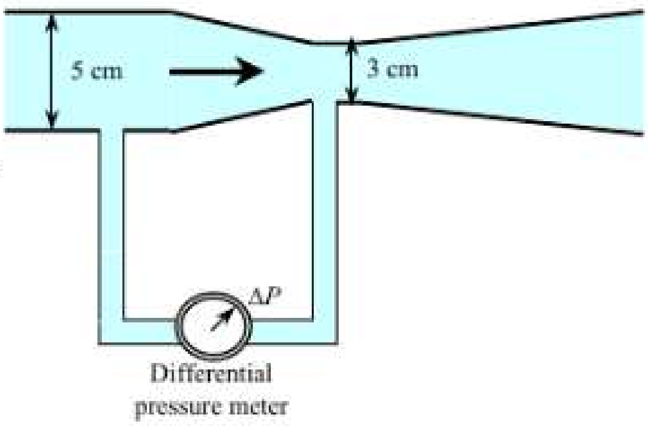
8–103 Diferansiyel basınç ölçer ile tertibatlandırılmış bir Venturimetre, 5 cm çaplı yatay bir borudaki 15°C'deki suyun (ρ = 999.1 kg/m^3) debisini ölçmek için kullanılmaktadır. Venturimetre boynunun çapı 3 cm ve ölçülen basınç düşüşü 5 kPa'dır. Debi katsayısını 0.98 alarak suyun hacimsel debisini ve borudaki ortalama hızı hesaplayınız. *Cevaplar:* 2.35 L/s ve 1.20 m/s

ŞEKİL P8–103

8-103 A Venturi meter equipped with a differential pressure gage is used to measure to flow rate of water through a horizontal pipe. For a given pressure drop, the volume flow rate of water and the average velocity through the pipe are to be determined.

Assumptions The flow is steady and incompressible.

Properties The density of water is given to be ρ = 999.1 kg/m^3. The discharge coefficient of Venturi meter is given to be C_d = 0.98.



Analysis The diameter ratio and the throat area of the meter are

$$\beta = d / D = 3/5 = 0.60$$

$$A_0 = \pi d^2/4 = \pi(0.03\,\text{m})^2/4 = 7.069\times10^{-4}\ \text{m}^2$$

Noting that ΔP = 5 kPa = 5000 N/m^2, the flow rate becomes

$$\dot{V} = A_o C_d \sqrt{\frac{2(P_1 - P_2)}{\rho(1-\beta^4)}}$$

$$= (7.069\times10^{-4}\ \text{m}^2)(0.98)\sqrt{\frac{2\times5000\ \text{N/m}^2}{(999.1\ \text{kg/m}^3)((1-0.60^4)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)}$$

$$= \mathbf{0.00235\ m^3/s}$$

which is equivalent to 2.35 L/s. The average flow velocity in the pipe is determined by dividing the flow rate by the cross-sectional area of the pipe,

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.00235\ \text{m}^3/\text{s}}{\pi(0.05\,\text{m})^2/4} = \mathbf{1.20\ m/s}$$

Discussion Note that the flow rate is proportional to the square root of pressure difference across the Venturi meter.