Solutions Manual for

Fluid Mechanics: Fundamentals and Applications

Third Edition

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CHAPTER 5
BERNOULLI AND ENERGY EQUATIONS

Conservation of Mass

5-1C

Solution We are to name some conserved and non-conserved quantities.

Analysis **Mass, energy, momentum, and electric charge are conserved**, and **volume and entropy are not conserved** during a process.

Discussion Students may think of other answers that may be equally valid.

5-2C

Solution We are to discuss mass and volume flow rates and their relationship.

Analysis *Mass flow rate* is the **amount of mass flowing through a cross-section per unit time** whereas *volume flow rate* is the **amount of volume flowing through a cross-section per unit time**.

Discussion Mass flow rate has dimensions of mass/time while volume flow rate has dimensions of volume/time.

5-3C

Solution We are to discuss the mass flow rate entering and leaving a control volume.

Analysis The amount of mass or energy entering a control volume **does not have to be equal** to the amount of mass or energy leaving during an unsteady-flow process.

Discussion If the process is steady, however, the two mass flow rates must be equal; otherwise the amount of mass would have to increase or decrease inside the control volume, which would make it unsteady.

5-4C

Solution We are to discuss steady flow through a control volume.

Analysis Flow through a control volume is *steady* when it involves **no changes with time at any specified position**.

Discussion This applies to any variable we might consider – pressure, velocity, density, temperature, etc.

5-5C

Solution We are to discuss whether the flow is steady through a given control volume.

Analysis **No**, a flow with the same volume flow rate at the inlet and the exit is **not necessarily steady** (even if the density is constant – see Discussion). ***To be steady, the mass flow rate through the device must remain constant in time, and no variables can change with time at any specified spatial position***.

Discussion If the question had stated that the two *mass* flow rates were equal, then the answer would still be *not necessarily*. As a counter-example, consider the steadily increasing flow of an incompressible liquid through the device. At any instant in time, the mass flow rate in must equal the mass flow rate out since there is nowhere else for the liquid to go. However, the mass flow rate itself is changing with time, and hence the problem is unsteady. Can you think of another counter-example?

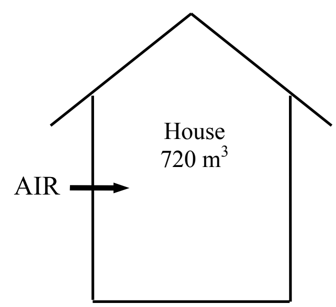
5-6

Solution A house is to be cooled by drawing in cool night time air continuously. For a specified air exchange rate, the required flow rate of the fan and the average discharge speed of air are to be determined.

Assumptions Flow through the fan is steady.

Analysis The volume of the house is given to be $V_{\text{house}} = 720\ \text{m}^3$. Noting that this volume of air is to be replaced every $\Delta t = 20\ \text{min}$, the required volume flow rate of air is

$$\dot{V} = \frac{V_{\text{room}}}{\Delta t} = \frac{720\ \text{m}^3}{20\ \text{min}}\left(\frac{1\ \text{min}}{60\ \text{s}}\right) = \mathbf{0.60\ m^3/s}$$



For the given fan diameter, the average discharge speed is determined to be

$$V = \frac{\dot{V}}{A_c} = \frac{\dot{V}}{\pi D^2/4} = \frac{0.60\ \text{m}^3/\text{s}}{\pi(0.5\ \text{m})^2/4} = \mathbf{3.06\ m/s}$$

Discussion Note that the air velocity and thus the noise level is low because of the large fan diameter.

5-7E

Solution A garden hose is used to fill a water bucket. The volume and mass flow rates of water, the filling time, and the discharge velocity are to be determined.

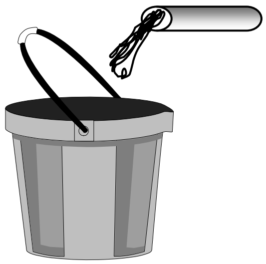
Assumptions **1** Water is an incompressible substance. **2** Flow through the hose is steady. **3** There is no waste of water by splashing.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis (*a*) The volume and mass flow rates of water are

$$\dot{V} = AV = (\pi D^2/4)V = [\pi(1/12\text{ ft})^2/4](8\text{ ft/s}) = 0.04363\text{ ft}^3\text{/s} \cong \mathbf{0.0436\ ft^3/s}$$

$$\dot{m} = \rho\dot{V} = (62.4\text{ lbm/ft}^3)(0.04363\text{ ft}^3\text{/s}) = \mathbf{2.72\ lbm/s}$$



(*b*) The time it takes to fill a 20-gallon bucket is

$$\Delta t = \frac{V}{\dot{V}} = \frac{20\text{ gal}}{0.04363\text{ ft}^3\text{/s}}\left(\frac{1\text{ ft}^3}{7.4804\text{ gal}}\right) = \mathbf{61.3\ s}$$

(*c*) The average discharge velocity of water at the nozzle exit is

$$V_e = \frac{\dot{V}}{A_e} = \frac{\dot{V}}{\pi D_e^2/4} = \frac{0.04363\text{ ft}^3\text{/s}}{[\pi(0.5/12\text{ ft})^2/4]} = \mathbf{32\ ft/s}$$

Discussion Note that for a given flow rate, the average velocity is inversely proportional to the square of the velocity. Therefore, when the diameter is reduced by half, the velocity quadruples.

5-8E

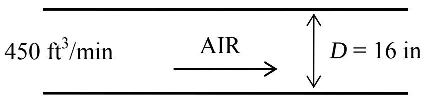
Solution The ducts of an air-conditioning system pass through an open area. The inlet velocity and the mass flow rate of air are to be determined.

Assumptions Flow through the air conditioning duct is steady.

Properties The density of air is given to be 0.082 lbm/ft^3 at the inlet.

Analysis The inlet velocity of air and the mass flow rate through the duct are

$$V_1 = \frac{\dot{V}_1}{A_1} = \frac{\dot{V}_1}{\pi D^2/4} = \frac{450\text{ ft}^3\text{/min}}{\pi(16/12\text{ ft})^2/4} = \mathbf{322\ ft/min} = \mathbf{26.9\ ft/s}$$



$$\dot{m} = \rho_1\dot{V}_1 = (0.082\text{ lbm/ft}^3)(450\text{ ft}^3/\text{min}) = 36.9\text{ lbm/min} = \mathbf{0.615\ lbm/s}$$

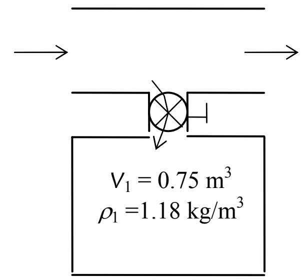
Discussion The mass flow rate though a duct must remain constant in steady flow; however, the volume flow rate varies since the density varies with the temperature and pressure in the duct.

5-9

Solution A rigid tank initially contains air at atmospheric conditions. The tank is connected to a supply line, and air is allowed to enter the tank until the density rises to a specified level. The mass of air that entered the tank is to be determined.

Properties The density of air is given to be 1.18 kg/m^3 at the beginning, and 4.95 kg/m^3 at the end.

Analysis We take the tank as the system, which is a control volume since mass crosses the boundary. The mass balance for this system can be expressed as



Mass balance: $m_{in} - m_{out} = \Delta m_{\text{system}} \rightarrow m_i = m_2 - m_1 = \rho_2 V - \rho_1 V$

Substituting, $m_i = (\rho_2 - \rho_1)V = [(4.95 - 1.18)\text{ kg/m}^3](0.75\text{ m}^3) = \mathbf{2.83\ kg}$

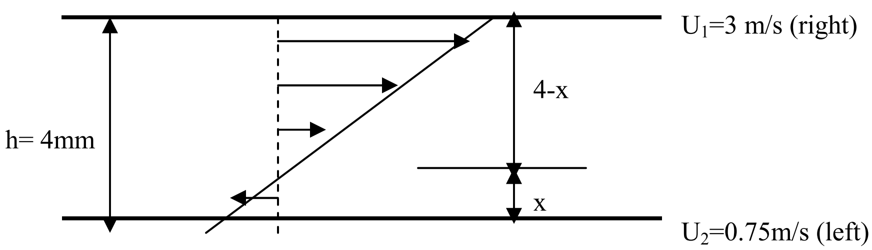
Therefore, **2.83 kg of mass entered the tank**.

Discussion Tank temperature and pressure do not enter into the calculations.

5-10

Solution A Newtonian fluid flows between two parallel plates. The upper plate moves to right and bottom one moves to the left. The net flow rate is to be determined.

Analysis From the similarity of the triangles we write



$$\frac{4-x}{x} = \frac{3}{0.75}$$

$3x = (4\text{-}x)(0.75)$

$3x = 3 - 0.75x$

$x = 0.8$ mm

$y = 4 - x = 3.2$ mm

$$\dot{V}_{net} = (3.2\times10^{-3})(5\times10^{-2})\frac{3}{2} - (0.8\times10^{-3})(5\times10^{-2})\frac{0.75}{2}$$

$$\dot{V}_{net} = 24\times10^{-6} - 15\times10^{-6} = \mathbf{9\times10^{-6}\ cm^3/s}$$

5-11

Solution Water is pumped out pf a fully-filled semi-circular cross section tank. The time needed to drop the water level to a specified value is to be determined in terms of given parameters.

Analysis From the conservation of mass, we write

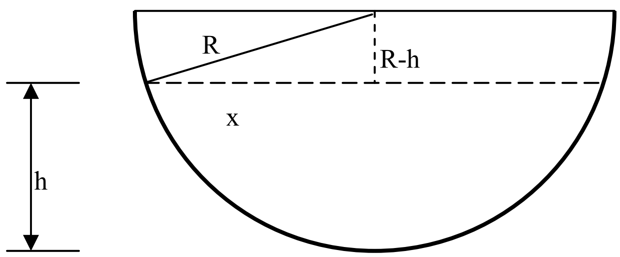
$$Qdt = -A_t dh$$

or

$$dt = -\frac{A_T}{Q}dh = -\frac{\pi x^2}{Kh^2}dh = -\frac{\pi\left[R^2-(R-h)^2\right]}{Kh^2}dh = -\frac{\pi}{K}\frac{2R-h}{h}dh$$

Integrating from h_1=R to h_2=0, we get

$$t = \frac{\pi}{K}\left(h - 2R\ln(h)\right)_R^H = -\frac{\pi}{K}\left[(R-H) + \ln\left(\frac{H}{R}\right)^{2R}\right]$$



5-12

Solution A desktop computer is to be cooled by a fan at a high elevation where the air density is low. The mass flow rate of air through the fan and the diameter of the casing for a given velocity are to be determined.

Assumptions Flow through the fan is steady.

Properties The density of air at a high elevation is given to be 0.7 kg/m^3.

Analysis The mass flow rate of air is

$$\dot{m}_{air} = \rho \dot{V}_{air} = (0.700 \text{ kg/m}^3)(0.400 \text{ m}^3\text{/min}) = 0.280 \text{ kg/min} = \mathbf{0.00467 \text{ kg/s}}$$

If the mean velocity is 110 m/min, the diameter of the casing is

$$\dot{V} = AV = \frac{\pi D^2}{4}V \quad \rightarrow \quad D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.400 \text{ m}^3\text{/min})}{\pi(110 \text{ m/min})}} = \mathbf{0.068 \text{ m}}$$

Therefore, the diameter of the casing must be **at least 5.69 cm** to ensure that the mean velocity does not exceed 110 m/min.

Discussion This problem shows that engineering systems are sized to satisfy given imposed constraints.

5-13

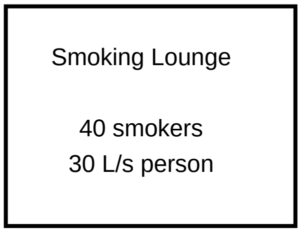
Solution A smoking lounge that can accommodate 40 smokers is considered. The required minimum flow rate of air that needs to be supplied to the lounge and the diameter of the duct are to be determined.

Assumptions Infiltration of air into the smoking lounge is negligible.

Properties The minimum fresh air requirements for a smoking lounge is given to be 30 L/s per person.

Analysis The required minimum flow rate of air that needs to be supplied to the lounge is determined directly from

$$\dot{V}_{\text{air}} = \dot{V}_{\text{air per person}}\,(\text{No. of persons})$$
$$= (30\ \text{L/s}\cdot\text{person})(40\ \text{persons}) = 1200\ \text{L/s} = \mathbf{1.2\ m^3/s}$$



The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2/4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(1.2\ \text{m}^3/\text{s})}{\pi(8\ \text{m/s})}} = \mathbf{0.437\ m}$$

Therefore, the diameter of the fresh air duct should be **at least 43.7 cm** if the velocity of air is not to exceed 8 m/s.

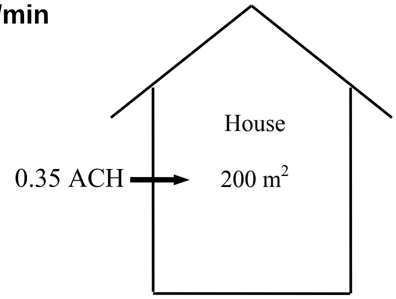
Discussion Fresh air requirements in buildings must be taken seriously to avoid health problems.

5-14

Solution The minimum fresh air requirements of a residential building is specified to be 0.35 air changes per hour. The size of the fan that needs to be installed and the diameter of the duct are to be determined.

Analysis The volume of the building and the required minimum volume flow rate of fresh air are

$$V_{\text{room}} = (2.7\ \text{m})(200\ \text{m}^2) = 540\ \text{m}^3$$
$$\dot{V} = V_{\text{room}} \times \text{ACH} = (540\ \text{m}^3)(0.35/\text{h}) = 189\ \text{m}^3/\text{h} = 189{,}000\ \text{L/h} = \mathbf{3150\ L/min}$$



The volume flow rate of fresh air can be expressed as

$$\dot{V} = VA = V(\pi D^2/4)$$

Solving for the diameter D and substituting,

$$D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(189/3600\ \text{m}^3/\text{s})}{\pi(5\ \text{m/s})}} = \mathbf{0.116\ m}$$

Therefore, the diameter of the fresh air duct should be **at least 11.6 cm** if the velocity of air is not to exceed 5 m/s.

Discussion Fresh air requirements in buildings must be taken seriously to avoid health problems.

5-15

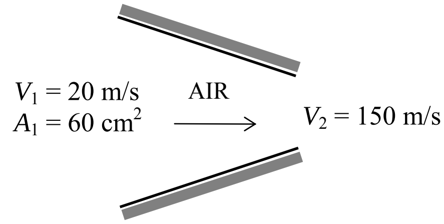
Solution Air is accelerated in a nozzle. The mass flow rate and the exit area of the nozzle are to be determined.

Assumptions Flow through the nozzle is steady.

Properties The density of air is given to be 2.21 kg/m³ at the inlet, and 0.762 kg/m³ at the exit.

Analysis (*a*) The mass flow rate of air is determined from the inlet conditions to be

$$\dot{m} = \rho_1 A_1 V_1 = (2.21 \text{ kg/m}^3)(0.006 \text{ m}^2)(20 \text{ m/s}) = 0.2652 \text{ kg/s} \cong \mathbf{0.265 \text{ kg/s}}$$



(*b*) There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$.

Then the exit area of the nozzle is determined to be

$$\dot{m} = \rho_2 A_2 V_2 \longrightarrow A_2 = \frac{\dot{m}}{\rho_2 V_2} = \frac{0.2652 \text{ kg/s}}{(0.762 \text{ kg/m}^3)(150 \text{ m/s})} = 0.00232 \text{ m}^2 = \mathbf{23.2 \text{ cm}^2}$$

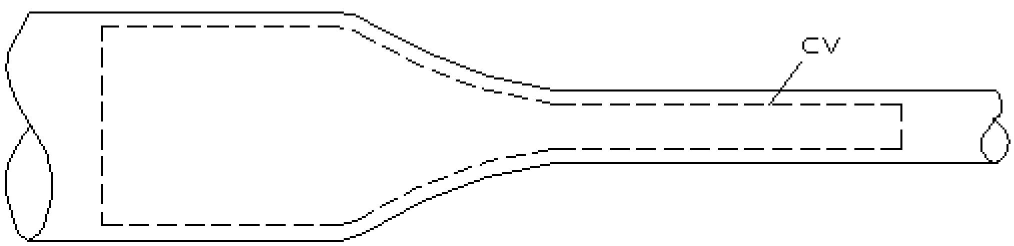
Discussion Since this is a compressible flow, we must equate mass flow rates, not volume flow rates.

5-16

Solution Air flows in a varying crosss section pipe. The speed at a specified section is to be determined.

Assumptions Flow through the pipe is steady.

Analysis



Applying conservation of mass for the cv shown,

$$\frac{\partial}{\partial t}\int_{cv} \rho . d\forall + \int_{cs} \rho . \vec{V} . \vec{n} . dA = 0 \quad , \quad -\rho . V_1 . A_1 + \rho_2 . A_2 . V_2 = 0 \qquad \overline{V_1} = \frac{\rho_2 . A_2 . V_2}{\rho_1 . A_1} \ ,$$

$$\rho = \frac{P_{abs}}{RT} \quad , \quad \rho_1 = \frac{P_{1(abs)}}{RT_1} \quad , \quad \rho_2 = \frac{P_{2(abs)}}{RT_2}$$

$$\frac{\rho_2}{\rho_1} = \frac{P_{2(abs)}}{P_{1(abs)}} \quad , \quad \frac{A_2}{A_1} = \frac{\frac{\pi d^2}{4}}{\frac{\pi D^2}{4}} = \left(\frac{d}{D}\right)^2$$

$$\overline{V_1} = \frac{110}{150} . \left(\frac{1}{3}\right)^2 .30 = \mathbf{2.44 \ \ m/s}$$

5-17

Solution Air is expanded and is accelerated as it is heated by a hair dryer of constant diameter. The percent increase in the velocity of air as it flows through the drier is to be determined.

Assumptions Flow through the nozzle is steady.

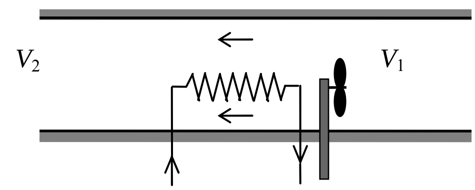
Properties The density of air is given to be 1.20 kg/m^3 at the inlet, and 1.05 kg/m^3 at the exit.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A V_1 = \rho_2 A V_2$$

$$\frac{V_2}{V_1} = \frac{\rho_1}{\rho_2} = \frac{1.20\ \text{kg/m}^3}{1.05\ \text{kg/m}^3} = 1.14 \qquad \text{(or, an increase of } \mathbf{14\%}\text{)}$$



Therefore, the air velocity increases **14%** as it flows through the hair drier.

Discussion It makes sense that the velocity *increases* since the density *decreases*, but the mass flow rate is constant.

Mechanical Energy and Efficiency

5-18C

Solution We are to define and discuss turbine, generator, and turbine-generator efficiency.

Analysis Turbine efficiency, generator efficiency, and *combined turbine-generator efficiency* are defined as follows:

$$\eta_{\text{turbine}} = \frac{\text{Mechanical energy output}}{\text{Mechanical energy extracted from the fluid}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|}$$

$$\eta_{\text{generator}} = \frac{\text{Electrical power output}}{\text{Mechanical power input}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{W}_{\text{shaft,in}}}$$

$$\eta_{\text{turbine-gen}} = \eta_{\text{turbine}}\eta_{\text{generaor}} = \frac{\dot{W}_{\text{elect,out}}}{\dot{E}_{\text{mech,in}} - \dot{E}_{\text{mech,out}}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|}$$

Discussion Most turbines are connected directly to a generator, so the combined efficiency is a useful parameter.

5-19C

Solution We are to define and discuss mechanical efficiency.

Analysis *Mechanical efficiency* is defined as **the ratio of the mechanical energy output to the mechanical energy input**. A mechanical efficiency of 100% for a hydraulic turbine means that the entire mechanical energy of the fluid is converted to mechanical (shaft) work.

Discussion No real fluid machine is 100% efficient, due to frictional losses, etc. – the second law of thermodynamics.

5-20C

Solution We are to define and discuss pump-motor efficiency.

Analysis The *combined pump-motor efficiency* of a pump/motor system is defined as the ratio of the increase in the mechanical energy of the fluid to the electrical power consumption of the motor,

$$\eta_{\text{pump-motor}} = \eta_{\text{pump}}\eta_{\text{motor}} = \frac{\dot{E}_{\text{mech,out}} - \dot{E}_{\text{mech,in}}}{\dot{W}_{\text{elect,in}}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{\dot{W}_{\text{pump}}}{\dot{W}_{\text{elect,in}}}$$

The combined pump-motor efficiency cannot be greater than either of the pump or motor efficiency since both pump and motor efficiencies are less than 1, and the product of two numbers that are less than one is less than either of the numbers.

Discussion Since many pumps are supplied with an integrated motor, pump-motor efficiency is a useful parameter.

5-21C

Solution We are to discuss mechanical energy and how it differs from thermal energy.

Analysis *Mechanical energy* is the **form of energy that can be converted to mechanical work completely and directly by a mechanical device** such as a propeller. It differs from thermal energy in that **thermal energy cannot be converted to work directly and completely**. The forms of mechanical energy of a fluid stream are kinetic, potential, and flow energies.

Discussion It would be nice if we could convert thermal energy completely into work. However, this would violate the second law of thermodynamics.

5-22

Solution Wind is blowing steadily at a certain velocity. The mechanical energy of air per unit mass, the power generation potential, and the actual electric power generation are to be determined.

Assumptions **1** The wind is blowing steadily at a constant uniform velocity. **2** The efficiency of the wind turbine is independent of the wind speed.

Properties The density of air is given to be $\rho = 1.25$ kg/m^3.

Analysis Kinetic energy is the only form of mechanical energy the wind possesses, and it can be converted to work entirely. Therefore, the power potential of the wind is its kinetic energy, which is $V^2/2$ per unit mass, and $\dot{m}V^2/2$ for a given mass flow rate:

$$e_{\text{mech}} = ke = \frac{V^2}{2} = \frac{(8\text{ m/s})^2}{2}\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.032\text{ kJ/kg}$$

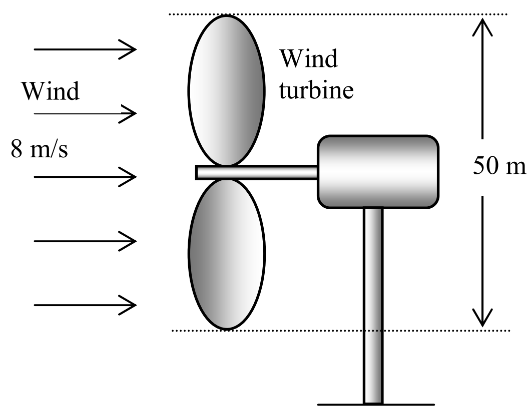
$$\dot{m} = \rho V A = \rho V\frac{\pi D^2}{4} = (1.25\text{ kg/m}^3)(8\text{ m/s})\frac{\pi(50\text{ m})^2}{4} = 19{,}635\text{ kg/s}$$

$$\dot{W}_{\max} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (19{,}635\text{ kg/s})(0.032\text{ kJ/kg}) = \mathbf{628\text{ kW}}$$

The actual electric power generation is determined by multiplying the power generation potential by the efficiency,

$$\dot{W}_{\text{elect}} = \eta_{\text{wind turbine}}\dot{W}_{\max} = (0.30)(628\text{ kW}) = \mathbf{188\text{ kW}}$$

Therefore, 283 kW of actual power can be generated by this wind turbine at the stated conditions.



Discussion The power generation of a wind turbine is proportional to the cube of the wind velocity, and thus the power generation will change strongly with the wind conditions.

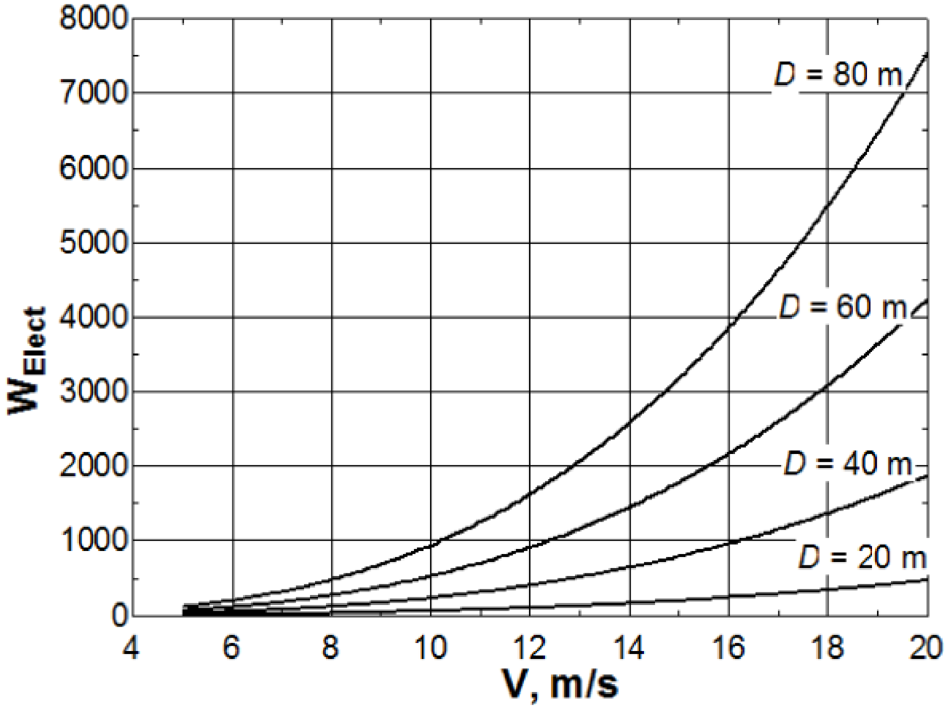
5-23

Solution The previous problem is reconsidered. The effect of wind velocity and the blade span diameter on wind power generation as the velocity varies from 5 m/s to 20 m/s in increments of 5 m/s, and the diameter varies from 20 m to 80 m in increments of 20 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
D1=20 "m"
D2=40 "m"
D3=60 "m"
D4=80 "m"
Eta=0.30
rho=1.25 "kg/m3"
m1_dot=rho*V*(pi*D1^2/4); W1_Elect=Eta*m1_dot*(V^2/2)/1000 "kW"
m2_dot=rho*V*(pi*D2^2/4); W2_Elect=Eta*m2_dot*(V^2/2)/1000 "kW"
m3_dot=rho*V*(pi*D3^2/4); W3_Elect=Eta*m3_dot*(V^2/2)/1000 "kW"
m4_dot=rho*V*(pi*D4^2/4); W4_Elect=Eta*m4_dot*(V^2/2)/1000 "kW"
```

D, m	*V*, m/s	*m*, kg/s	W_{elect}, kW
20	5	1,963	7
	10	3,927	59
	15	5,890	199
	20	7,854	471
40	5	7,854	29
	10	15,708	236
	15	23,562	795
	20	31,416	1885
60	5	17,671	66
	10	35,343	530
	15	53,014	1789
	20	70,686	4241
80	5	31,416	118
	10	62,832	942
	15	94,248	3181
	20	125,664	7540



Discussion Wind turbine power output is obviously nonlinear with respect to both velocity and diameter.

5-24E

Solution A differential thermocouple indicates that the temperature of water rises a certain amount as it flows through a pump at a specified rate. The mechanical efficiency of the pump is to be determined.

Assumptions **1** The pump is adiabatic so that there is no heat transfer with the surroundings, and the temperature rise of water is completely due to frictional heating. **2** Water is an incompressible substance.

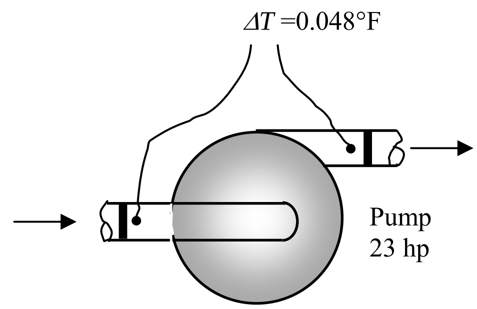
Properties We take the density of water to be $\rho = 62.4$ lbm/ft^3 and its specific heat to be $c = 1.0$ Btu/lbm·°F.

Analysis The increase in the temperature of water is due to the conversion of mechanical energy to thermal energy, and the amount of mechanical energy converted to thermal energy is equal to the increase in the internal energy of water,

$$\dot{m} = \rho\dot{V} = (62.4\ \text{lbm/ft}^3)(1.5\ \text{ft}^3/\text{s}) = 93.6\ \text{lbm/s}$$

$$\dot{E}_{\text{mech, loss}} = \Delta\dot{U} = \dot{m}c\Delta T$$

$$= (93.6\ \text{lbm/s})(1.0\ \text{Btu/lbm}\cdot{}^\circ\text{F})(0.048^\circ\text{F})\left(\frac{1\ \text{hp}}{0.7068\ \text{Btu/s}}\right) = 6.36\ \text{hp}$$



The mechanical efficiency of the pump is determined from the general definition of mechanical efficiency,

$$\eta_{\text{pump}} = 1 - \frac{\dot{E}_{\text{mech,loss}}}{\dot{W}_{\text{mech, in}}} = 1 - \frac{6.36\ \text{hp}}{23\ \text{hp}} = 0.724 \quad \text{or} \quad \mathbf{72.4\%}$$

Discussion Note that despite the conversion of more than one-third of the mechanical power input into thermal energy, the temperature of water rises by only a small fraction of a degree. Therefore, the temperature rise of a fluid due to frictional heating is usually negligible in heat transfer analysis.

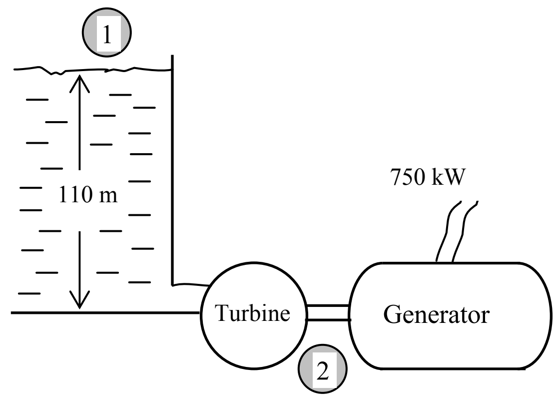
5-25

Solution A hydraulic turbine-generator is generating electricity from the water of a large reservoir. The combined turbine-generator efficiency and the turbine efficiency are to be determined.

Assumptions **1** The elevation of the reservoir remains constant. **2** The mechanical energy of water at the turbine exit is negligible.

Analysis We take the free surface of the reservoir to be point 1 and the turbine exit to be point 2. We also take the turbine exit as the reference level ($z_2 = 0$), and thus the potential energy at points 1 and 2 are $pe_1 = gz_1$ and $pe_2 = 0$. The flow energy P/ρ at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($ke_1 = ke_2 = 0$) since the water at point 1 is essentially motionless, and the kinetic energy of water at turbine exit is assumed to be negligible. The potential energy of water at point 1 is

$$pe_1 = gz_1 = (9.81\ \text{m/s}^2)(110\ \text{m})\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2}\right) = 1.079\ \text{kJ/kg}$$



Then the rate at which the mechanical energy of the fluid is supplied to the turbine become

$$\left|\Delta \dot{E}_{\text{mech,fluid}}\right| = \dot{m}(e_{\text{mech,in}} - e_{\text{mech,out}}) = \dot{m}(pe_1 - 0) = \dot{m}pe_1$$
$$= (900\ \text{kg/s})(1.079\ \text{kJ/kg})$$
$$= 971.2\ \text{kW}$$

The combined turbine-generator and the turbine efficiency are determined from their definitions,

$$\eta_{\text{turbine-gen}} = \frac{\dot{W}_{\text{elect,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|} = \frac{750\ \text{kW}}{971.2\ \text{kW}} = 0.772 \quad \text{or} \quad \mathbf{77.2\%}$$

$$\eta_{\text{turbine}} = \frac{\dot{W}_{\text{shaft,out}}}{|\Delta \dot{E}_{\text{mech,fluid}}|} = \frac{800\ \text{kW}}{971.2\ \text{kW}} = 0.824 \quad \text{or} \quad \mathbf{82.4\%}$$

Therefore, the reservoir supplies 971.2 kW of mechanical energy to the turbine, which converts 800 kW of it to shaft work that drives the generator, which generates 750 kW of electric power.

Discussion This problem can also be solved by taking point 1 to be at the turbine inlet, and using flow energy instead of potential energy. It would give the same result since the flow energy at the turbine inlet is equal to the potential energy at the free surface of the reservoir.

5-26

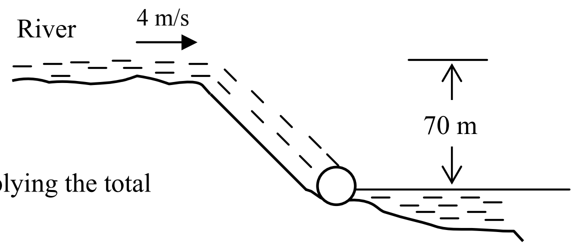
Solution A river is flowing at a specified velocity, flow rate, and elevation. The total mechanical energy of the river water per unit mass, and the power generation potential of the entire river are to be determined.

Assumptions **1** The elevation given is the elevation of the free surface of the river. **2** The velocity given is the average velocity. **3** The mechanical energy of water at the turbine exit is negligible.

Properties We take the density of water to be $\rho = 1000\ \text{kg/m}^3$.

Analysis Noting that the sum of the flow energy and the potential energy is constant for a given fluid body, we can take the elevation of the entire river water to be the elevation of the free surface, and ignore the flow energy. Then the total mechanical energy of the river water per unit mass becomes

$$e_{\text{mech}} = pe + ke = gh + \frac{V^2}{2}$$
$$= \left((9.81\ \text{m/s}^2)(70\ \text{m}) + \frac{(4\ \text{m/s})^2}{2} \right)\left(\frac{1\ \text{kJ/kg}}{1000\ \text{m}^2/\text{s}^2} \right)$$
$$= 0.695\ \text{kJ/kg}$$



The power generation potential of the river water is obtained by multiplying the total mechanical energy by the mass flow rate,

$$\dot{m} = \rho \dot{V} = (1000\ \text{kg/m}^3)(500\ \text{m}^3/\text{s}) = 500{,}000\ \text{kg/s}$$

$$\dot{W}_{\text{max}} = \dot{E}_{\text{mech}} = \dot{m}e_{\text{mech}} = (500{,}000\ \text{kg/s})(0.695\ \text{kg/s}) = 347{,}350\ \text{kW} \cong \mathbf{347\ MW}$$

Therefore, 347 MW of power can be generated from this river as it discharges into the lake if its power potential can be recovered completely.

Discussion Note that the kinetic energy of water is negligible compared to the potential energy, and it can be ignored in the analysis. Also, the power output of an actual turbine will be less than 347 MW because of losses and inefficiencies.

5-27

Solution Water is pumped from a lake to a storage tank at a specified rate. The overall efficiency of the pump-motor unit and the pressure difference between the inlet and the exit of the pump are to be determined.

Assumptions **1** The elevations of the tank and the lake remain constant. **2** Frictional losses in the pipes are negligible. **3** The changes in kinetic energy are negligible. **4** The elevation difference across the pump is negligible.

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

Analysis (*a*) We take the free surface of the lake to be point 1 and the free surfaces of the storage tank to be point 2. We also take the lake surface as the reference level ($z_1 = 0$), and thus the potential energy at points 1 and 2 are $pe_1 = 0$ and $pe_2 = gz_2$. The flow energy at both points is zero since both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$). Further, the kinetic energy at both points is zero ($ke_1 = ke_2 = 0$) since the water at both locations is essentially stationary. The mass flow rate of water and its potential energy at point 2 are

$$\dot{m} = \rho\dot{V} = (1000\text{ kg/m}^3)(0.070\text{ m}^3/\text{s}) = 70\text{ kg/s}$$

$$pe_1 = gz_1 = (9.81\text{ m/s}^2)(18\text{ m})\left(\frac{1\text{ kJ/kg}}{1000\text{ m}^2/\text{s}^2}\right) = 0.177\text{ kJ/kg}$$

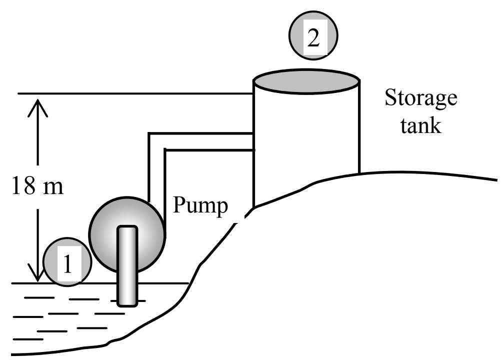
Then the rate of increase of the mechanical energy of water becomes

$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}(pe_2 - 0) = \dot{m}pe_2 = (70\text{ kg/s})(0.177\text{ kJ/kg}) = 12.4\text{ kW}$$

The overall efficiency of the combined pump-motor unit is determined from its definition,

$$\eta_{\text{pump-motor}} = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{W}_{\text{elect,in}}} = \frac{12.4\text{ kW}}{20.4\text{ kW}} = 0.606 \quad \text{or} \quad \mathbf{60.6\%}$$

(*b*) Now we consider the pump. The change in the mechanical energy of water as it flows through the pump consists of the change in the flow energy only since the elevation difference across the pump and the change in the kinetic energy are negligible. Also, this change must be equal to the useful mechanical energy supplied by the pump, which is 12.4 kW:



$$\Delta\dot{E}_{\text{mech,fluid}} = \dot{m}(e_{\text{mech,out}} - e_{\text{mech,in}}) = \dot{m}\frac{P_2 - P_1}{\rho} = \dot{V}\Delta P$$

Solving for ΔP and substituting,

$$\Delta P = \frac{\Delta\dot{E}_{\text{mech,fluid}}}{\dot{V}} = \frac{12.4\text{ kJ/s}}{0.070\text{ m}^3/\text{s}}\left(\frac{1\text{ kPa}\cdot\text{m}^3}{1\text{ kJ}}\right) = \mathbf{177\ kPa}$$

Therefore, the pump must boost the pressure of water by 177 kPa in order to raise its elevation by 18 m.

Discussion Note that only two-thirds of the electric energy consumed by the pump-motor is converted to the mechanical energy of water; the remaining one-third is wasted because of the inefficiencies of the pump and the motor.

Bernoulli Equation

5-28C

Solution We are to define stagnation pressure and discuss how it can be measured.

Analysis The **sum of the static and dynamic pressures** is called the *stagnation pressure*, and it is expressed as $\boxed{P_{\text{stag}} = P + \rho V^2 / 2}$. The stagnation pressure can be measured by **a Pitot tube whose inlet is normal to the flow**.

Discussion Stagnation pressure, as its name implies, is the pressure obtained when a flowing fluid is brought to rest isentropically, at a so-called *stagnation point*.

5-29C

Solution We are to express the Bernoulli equation in three different ways.

Analysis The Bernoulli equation is expressed in three different ways as follows:

(a) In terms of energies: $\boxed{\dfrac{P}{\rho} + \dfrac{V^2}{2} + gz = \text{constant}}$

(b) In terms of pressures: $\boxed{P + \rho\dfrac{V^2}{2} + \rho gz = \text{constant}}$

(c) in terms of heads: $\boxed{\dfrac{P}{\rho g} + \dfrac{V^2}{2g} + z = H = \text{constant}}$

Discussion You could, of course, express it in other ways, but these three are the most useful.

5-30C

Solution We are to discuss the three major assumptions used in the derivation of the Bernoulli equation.

Analysis The three major assumptions used in the derivation of the Bernoulli equation are that the flow is **steady**, **there is negligible frictional effects**, and the flow is **incompressible**.

Discussion If any one of these assumptions is not valid, the Bernoulli equation should not be used. Unfortunately, many people use it anyway, leading to errors.

5-31C

Solution We are to define and discuss static, dynamic, and hydrostatic pressure.

Analysis *Static pressure P* is the **actual pressure of the fluid**. *Dynamic pressure* $\rho V^2/2$ is the **pressure rise when the fluid in motion is brought to a stop isentropically**. *Hydrostatic pressure* ρgz is not pressure in a real sense since its value depends on the reference level selected, and it **accounts for the effects of fluid weight on pressure**. The sum of static, dynamic, and hydrostatic pressures is constant when flow is steady and incompressible, and when frictional effects are negligible.

Discussion The incompressible Bernoulli equation states that the sum of these three pressures is constant along a streamline; this approximation is valid only for steady and incompressible flow with negligible frictional effects.

5-32C

Solution We are to define streamwise acceleration and discuss how it differs from normal acceleration.

Analysis The **acceleration of a fluid particle along a streamline** is called *streamwise acceleration*, and it is due to a change in speed along a streamline. *Normal acceleration* (or centrifugal acceleration), on the other hand, is the **acceleration of a fluid particle in the direction normal to the streamline**, and it is due to a change in direction.

Discussion In a general fluid flow problem, both streamwise and normal acceleration are present.

5-33C

Solution We are to define and discuss pressure head, velocity head, and elevation head.

Analysis The *pressure head* $P/\rho g$ is the **height of a fluid column that produces the static pressure *P***. The *velocity head* $V^2/2$ is the **elevation needed for a fluid to reach the velocity *V* during frictionless free fall**. The *elevation head z* is the **height of a fluid relative to a reference level**.

Discussion It is often convenient in fluid mechanics to work with *head* – pressure expressed as an equivalent column height of fluid.

5-34C

Solution We are to explain how and why a siphon works, and its limitations.

Analysis Siphoning works because of the elevation and thus pressure difference between the inlet and exit of a tube. The pressure at the tube exit and at the free surface of a liquid is the atmospheric pressure. When the tube exit is below the free surface of the liquid, the elevation head difference drives the flow through the tube. At sea level, 1 atm pressure can support about 10.3 m of cold water (cold water has a low vapor pressure). Therefore, siphoning cold water over a 7 m wall is **theoretically feasible**.

Discussion In actual practice, siphoning is also limited by frictional effects in the tube, and by cavitation.

5-35C

Solution We are to discuss the hydraulic grade line in open-channel flow and at the outlet of a pipe.

Analysis For *open-channel flow*, the hydraulic grade line (HGL) **coincides with the free surface of the liquid**. At the exit of a pipe discharging to the atmosphere, HGL **coincides with the elevation of the pipe outlet**.

Discussion We are assuming incompressible flow, and the pressure at the pipe outlet is atmospheric.

5-36C

Solution We are to discuss the effect of liquid density on the operation of a siphon.

Analysis The lower density liquid can go over a higher wall, provided that cavitation pressure is not reached. Therefore, oil may be able to go over a higher wall than water.

Discussion However, frictional losses in the flow of oil in a pipe or tube are much greater than those of water since the viscosity of oil is much greater than that of water. When frictional losses are considered, the water may actually be able to be siphoned over a higher wall than the oil, depending on the tube diameter and length, etc.

5-37C

Solution We are to define hydraulic grade line and compare it to energy grade line.

Analysis **The curve that represents the sum of the static pressure and the elevation heads**, $P/\rho g + z$, is called the *hydraulic grade line* or HGL. The curve that represents the total head of the fluid, $P/\rho g + V^2/2g + z$, is called the *energy line* or EGL. Thus, in comparison, the energy grade line contains an extra kinetic-energy-type term. **For stationary bodies such as reservoirs or lakes, the EL and HGL coincide with the free surface of the liquid**.

Discussion The hydraulic grade line can rise or fall along flow in a pipe or duct as the cross-sectional area increases or decreases, whereas the energy grade line *always* decreases unless energy is added to the fluid (like with a pump).

5-38C

Solution We are to discuss and compare the operation of a manometer.

Analysis As the duct converges to a smaller cross-sectional area, the velocity increases. By Bernoulli's equation, the pressure therefore decreases. Thus **Manometer A** is correct since the pressure on the right side of the manometer is obviously smaller. According to the Bernoulli approximation, the fluid levels in the manometer are independent of the flow direction, and reversing the flow direction would have no effect on the manometer levels. **Manometer A is still correct if the flow is reversed**.

Discussion In reality, it is hard for a fluid to expand without the flow separating from the walls. Thus, reverse flow with such a sharp expansion would not produce as much of a pressure rise as that predicted by the Bernoulli approximation.

5-39C

Solution We are to discuss and compare two different types of manometer arrangements in a flow.

Analysis Arrangement 1 consists of a Pitot probe that measures the stagnation pressure at the pipe centerline, along with a static pressure tap that measures static pressure at the bottom of the pipe. Arrangement 2 is a Pitot-static probe that measures both stagnation pressure and static pressure at *nearly the same location* at the pipe centerline. Because of this, **arrangement 2 is more accurate**. However, it turns out that static pressure in a pipe varies with elevation across the pipe cross section in much the same way as in hydrostatics. Therefore, arrangement 1 is also very accurate, and the elevation difference between the Pitot probe and the static pressure tap is nearly compensated by the change in hydrostatic pressure. Since elevation changes are not important in either arrangement, there is **no change in our analysis when the water is replaced by air**.

Discussion Ignoring the effects of gravity, the pressure at the centerline of a turbulent pipe flow is actually somewhat smaller than that at the wall due to the turbulent eddies in the flow, but this effect is small.

5-40C

Solution We are to discuss the maximum rise of a jet of water from a tank.

Analysis With no losses and a 100% efficient nozzle, the water stream could reach to the water level in the tank, or 20 meters. In reality, friction losses in the hose, nozzle inefficiencies, orifice losses, and air drag would prevent attainment of the maximum theoretical height.

Discussion In fact, the actual maximum obtainable height is much smaller than this ideal theoretical limit.

5-41C

Solution We are to compare siphoning at sea level and on a mountain.

Analysis At sea level, a person can theoretically siphon water over a wall as high as 10.3 m. At the top of a high mountain where the pressure is about half of the atmospheric pressure at sea level, a person can theoretically siphon water over a wall that is only half as high. **An atmospheric pressure of 58.5 kPa is insufficient to support a 8.5 meter high siphon.**

Discussion In actual practice, siphoning is also limited by frictional effects in the tube, and by cavitation.

5-42

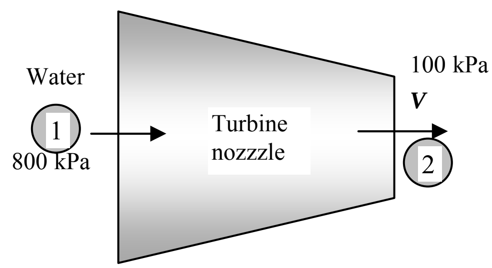
Solution In a power plant, water enters the nozzles of a hydraulic turbine at a specified pressure. The maximum velocity water can be accelerated to by the nozzles is to be determined.

Assumptions **1**The flow of water is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Water enters the nozzle with a low velocity.

Properties We take the density of water to be $\rho = 1000\ \text{kg/m}^3$.

Analysis We take points 1 and 2 at the inlet and exit of the nozzle, respectively. Noting that $V_1 \cong 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \to \quad \frac{P_1}{\rho g} = \frac{P_{atm}}{\rho g} + \frac{V_2^2}{2g} \quad \to \quad V_2 = \sqrt{\frac{2(P_1 - P_{atm})}{\rho}}$$



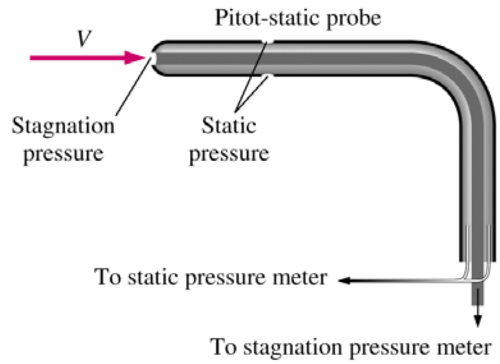
Substituting the given values, the nozzle exit velocity is determined to be

$$V_1 = \sqrt{\frac{2(800-100)\ \text{kPa}}{1000\ \text{kg/m}^3}\left(\frac{1000\ \text{N/m}^2}{1\ \text{kPa}}\right)\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)} = \mathbf{37.4\ m/s}$$

Discussion This is the maximum nozzle exit velocity, and the actual velocity will be less because of friction between water and the walls of the nozzle.

5-43

Solution The velocity of an aircraft is to be measured by a Pitot-static probe. For a given differential pressure reading, the velocity of the aircraft is to be determined.



Assumptions **1** The air flow over the aircraft is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Standard atmospheric conditions exist. **3** The wind effects are negligible.

Properties The density of the atmosphere at an elevation of 3000 m is ρ = 0.909 kg/m^3.

Analysis We take point 1 at the entrance of the tube whose opening is parallel to flow, and point 2 at the entrance of the tube whose entrance is normal to flow. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

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

Solving for V_1 and substituting,

$$V_1=\sqrt{\frac{2(P_{stag}-P_1)}{\rho}}=\sqrt{\frac{2(3000\text{ N/m}^2)}{0.909\text{ kg/m}^3}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}=\mathbf{81.2\text{ m/s}}=\mathbf{292\text{ km/h}}$$

since 1 Pa = 1 N/m^2 and 1 m/s = 3.6 km/h.

Discussion Note that the velocity of an aircraft can be determined by simply measuring the differential pressure on a Pitot-static probe.

5-44

Solution A Pitot-static probe is inserted into the duct of an air heating system parallel to flow, and the differential height of the water column is measured. The flow velocity and the pressure rise at the tip of the Pitot-static probe are to be determined.

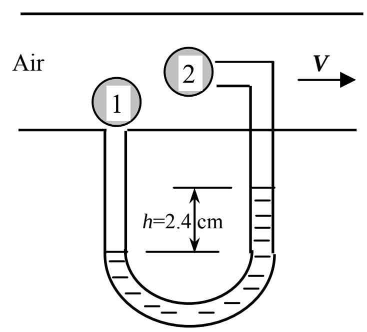
Assumptions **1** The flow through the duct is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Air is an ideal gas.

Properties We take the density of water to be ρ = 1000 kg/m³. The gas constant of air is R = 0.287 kPa·m³/kg·K.

Analysis We take point 1 on the side of the probe where the entrance is parallel to flow and is connected to the static arm of the Pitot-static probe, and point 2 at the tip of the probe where the entrance is normal to flow and is connected to the dynamic arm of the Pitot-static probe. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+\frac{V_1^2}{2g}=\frac{P_2}{\rho g} \quad\rightarrow\quad V=\sqrt{\frac{2(P_2-P_1)}{\rho_{air}}}$$

where the pressure rise at the tip of the Pitot-static probe is



$$P_2-P_1=\rho_w gh=(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.024\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)$$

$$=235\text{ N/m}^2=\mathbf{235\ Pa}$$

Also, $$\rho_{air}=\frac{P}{RT}=\frac{98\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(45+273\text{ K})}=1.074\text{ kg/m}^3$$

Substituting,

$$V_1=\sqrt{\frac{2(235\text{ N/m}^2)}{1.074\text{ kg/m}^3}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)}=\mathbf{20.9\ m/s}$$

Discussion Note that the flow velocity in a pipe or duct can be measured easily by a Pitot-static probe by inserting the probe into the pipe or duct parallel to flow, and reading the differential pressure height. Also note that this is the velocity at the location of the tube. Several readings at several locations in a cross-section may be required to determine the mean flow velocity.

5-45E

Solution The drinking water needs of an office are met by large water bottles with a plastic hose inserted in it. The minimum filling time of an 8-oz glass is to be determined when the bottle is full and when it is near empty.

Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** All losses are neglected to obtain the minimum filling time.

Analysis We take point 1 to be at the free surface of water in the bottle and point 2 at the exit of the tube so that $P_1 = P_2 = P_{atm}$ (the bottle is open to the atmosphere and water discharges into the atmosphere), $V_1 \cong 0$ (the bottle is large relative to the tube diameter), and $z_2 = 0$ (we take point 2 as the reference level). Then the Bernoulli equation simplifies to

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

Substituting, the discharge velocity of water and the filling time are determined as follows:

(*a*) *Full bottle* (z_1 = 3.5 ft):

$$V_2=\sqrt{2(32.2\text{ ft/s}^2)(3.5\text{ ft})}=15.0\text{ ft/s}$$

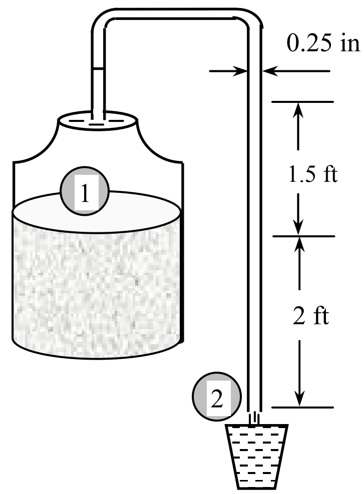
$$A=\pi D^2/4=\pi(0.25/12\text{ ft})^2/4=3.41\times10^{-4}\text{ ft}^2$$

$$\Delta t=\frac{V}{\dot V}=\frac{V}{AV_2}=\frac{0.00835\text{ ft}^3}{(3.41\times10^{-4}\text{ ft}^2)(15\text{ ft/s})}=\mathbf{1.6\ s}$$

(*b*) *Empty bottle* (z_1 = 2 ft):

$$V_2=\sqrt{2(32.2\text{ ft/s}^2)(2\text{ ft})}=11.3\text{ ft/s}$$

$$\Delta t=\frac{V}{\dot V}=\frac{V}{AV_2}=\frac{0.00835\text{ ft}^3}{(3.41\times10^{-4}\text{ ft}^2)(11.3\text{ ft/s})}=\mathbf{2.2\ s}$$



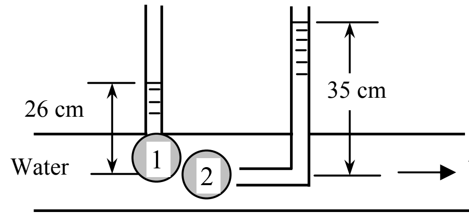
Discussion The siphoning time is determined assuming frictionless flow, and thus this is the *minimum time* required. In reality, the time will be longer because of friction between water and the tube surface.

5-46

Solution The static and stagnation pressures in a horizontal pipe are measured. The velocity at the center of the pipe is to be determined.

Assumptions The flow is steady, incompressible, and irrotational with negligible frictional effects in the short distance between the two pressure measurement locations (so that the Bernoulli equation is applicable).

Analysis We take points 1 and 2 along the centerline of the pipe, with point 1 directly under the piezometer and point 2 at the entrance of the Pitot-static probe (the stagnation point).



This is a steady flow with straight and parallel streamlines, and thus the static pressure at any point is equal to the hydrostatic pressure at that point. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

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

Substituting the P_1 and P_2 expressions give

$$\frac{V_1^2}{2g} = \frac{P_2 - P_1}{\rho g} = \frac{\rho g(h_{\text{pitot}} + R) - \rho g(h_{\text{piezo}} + R)}{\rho g} = \frac{\rho g(h_{\text{pitot}} - h_{\text{piezo}})}{\rho g} = h_{\text{pitot}} - h_{\text{piezo}}$$

Solving for V_1 and substituting,

$$V_1 = \sqrt{2g(h_{\text{pitot}} - h_{\text{piezo}})} = \sqrt{2(9.81\ \text{m/s}^2)[(0.35 - 0.26)\ \text{m}]} = \mathbf{1.33\ m/s}$$

Discussion Note that to determine the flow velocity, all we need is to measure the height of the excess fluid column in the Pitot-static probe.

5-47

Solution A water tank of diameter D_o and height H open to the atmosphere is initially filled with water. An orifice of diameter D with a smooth entrance (no losses) at the bottom drains to the atmosphere. Relations are to be developed for the time required for the tank to empty completely and half-way.

Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

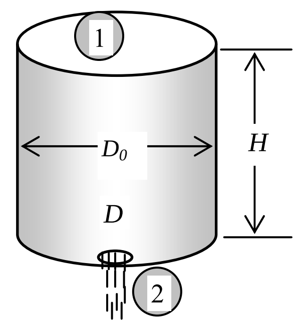
Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of orifice. We take the reference level at the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

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

For generality, we express the water height in the tank at any time t by z, and the discharge velocity by $V_2=\sqrt{2gz}$. Note that water surface in the tank moves down as the tank drains, and thus z is a variable whose value changes from H at the beginning to 0 when the tank is emptied completely.

We denote the diameter of the orifice by D, and the diameter of the tank by D_o. The flow rate of water from the tank is obtained by multiplying the discharge velocity by the orifice cross-sectional area,

$$\dot{V}=A_{\text{orifice}}V_2=\frac{\pi D^2}{4}\sqrt{2gz}$$



Then the amount of water that flows through the orifice during a differential time interval dt is

$$dV=\dot{V}dt=\frac{\pi D^2}{4}\sqrt{2gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV=A_{\text{tank}}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{2gz}\,dt=-\frac{\pi D_0^2}{4}dz \quad\to\quad dt=-\frac{D_0^2}{D^2}\sqrt{\frac{1}{2gz}}dz=-\frac{D_0^2}{D^2\sqrt{2g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_i = H$ to $t = t_f$ when $z = z_f$ gives

$$\int_{t=0}^{t_f}dt=-\frac{D_0^2}{D^2\sqrt{2g}}\int_{z=z_1}^{z_f}z^{-1/2}dz \quad\to\quad t_f=-\frac{D_0^2}{D^2\sqrt{2g}}\left.\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{z_f}=\frac{2D_0^2}{D^2\sqrt{2g}}\left(\sqrt{z_i}-\sqrt{z_f}\right)=\frac{D_0^2}{D^2}\left(\sqrt{\frac{2z_i}{g}}-\sqrt{\frac{2z_f}{g}}\right)$$

Then the discharging time for the two cases becomes as follows:

(*a*) *The tank empties halfway*: $z_i = H$ and $z_f = H/2$: $\boxed{t_f=\frac{D_0^2}{D^2}\left(\sqrt{\frac{2H}{g}}-\sqrt{\frac{H}{g}}\right)}$

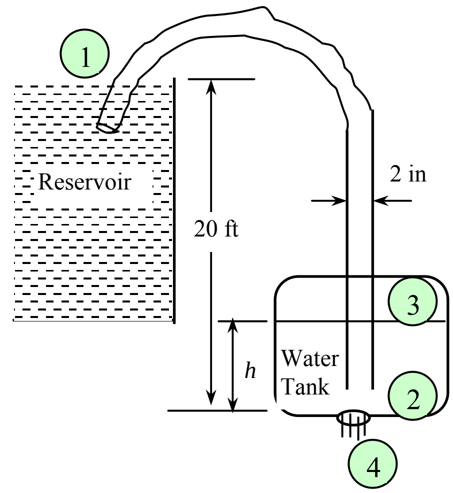
(*b*) *The tank empties completely*: $z_i = H$ and $z_f = 0$: $\boxed{t_f=\frac{D_0^2}{D^2}\sqrt{\frac{2H}{g}}}$

Discussion Note that the discharging time is inversely proportional to the square of the orifice diameter. Therefore, the discharging time can be reduced to one-fourth by doubling the diameter of the orifice.

5-48E

Solution A siphon pumps water from a large reservoir to a lower tank which is initially empty. Water leaves the tank through an orifice. The height the water will rise in the tank at equilibrium is to be determined.

Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Both the tank and the reservoir are open to the atmosphere. **3** The water level of the reservoir remains constant.



Analysis We take the reference level to be at the bottom of the tank, and the water height in the tank at any time to be h. We take point 1 to be at the free surface of reservoir, point 2 at the exit of the siphon, which is placed at the bottom of the tank, and point 3 at the free surface of the tank, and point 4 at the exit of the orifice at the bottom of the tank. Then z_1 = 20 ft, $z_2 = z_4 = 0$, $z_3 = h$, $P_1 = P_3 = P_4 = P_{atm}$ (the reservoir is open to the atmosphere and water discharges into the atmosphere) $P_2 = P_{atm}+\rho gh$ (the hydrostatic pressure at the bottom of the tank where the siphon discharges), and $V_1 \cong V_3 \cong 0$ (the free surfaces of reservoir and the tank are large relative to the tube diameter). Then the Bernoulli equation between 1-2 and 3-4 simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_{atm}}{\rho g}+z_1=\frac{P_{atm}+\rho gh}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1-2gh}=\sqrt{2g(z_1-h)}$$

$$\frac{P_3}{\rho g}+\frac{V_3^2}{2g}+z_3=\frac{P_4}{\rho g}+\frac{V_4^2}{2g}+z_4 \quad\rightarrow\quad h=\frac{V_4^2}{2g} \quad\rightarrow\quad V_4=\sqrt{2gh}$$

Noting that the diameters of the tube and the orifice are the same, the flow rates of water into and out of the tank will be the same when the water velocities in the tube and the orifice are equal since

$$\dot{V}_2=\dot{V}_4 \quad\rightarrow\quad AV_2=AV_4 \quad\rightarrow\quad V_2=V_4$$

Setting the two velocities equal to each other gives

$$V_2=V_4 \quad\rightarrow\quad \sqrt{2g(z_1-h)}=\sqrt{2gh} \quad\rightarrow\quad z_1-h=h \quad\rightarrow\quad h=\frac{z_1}{2}=\frac{20\text{ ft}}{2}=\mathbf{10\ ft}$$

Therefore, the water level in the tank will stabilize when the water level rises to 10 ft.

Discussion This result is obtained assuming negligible friction. The result would be somewhat different if the friction in the pipe and orifice were considered.

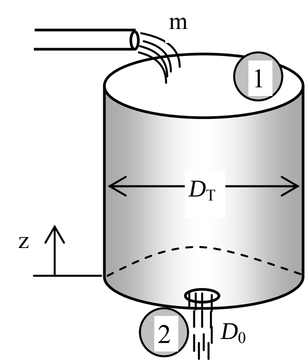
5-49

Solution Water enters an empty tank steadily at a specified rate. An orifice at the bottom allows water to escape. The maximum water level in the tank is to be determined, and a relation for water height *z* as a function of time is to be obtained.

Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow through the orifice is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Analysis (*a*) We take point 1 at the free surface of the tank, and point 2 at the exit of orifice. We take the reference level at the orifice ($z_2 = 0$), and take the positive direction of *z* to be upwards. 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 very low ($V_1 \cong 0$) (it becomes zero when the water in the tank reaches its maximum level), the Bernoulli equation between these two points simplifies to

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



Then the mass flow rate through the orifice for a water height of *z* becomes

$$\dot m_{out}=\rho\dot V_{out}=\rho A_{orifice}V_2=\rho\frac{\pi D_0^2}{4}\sqrt{2gz} \quad\rightarrow\quad z=\frac{1}{2g}\left(\frac{4\dot m_{out}}{\rho\pi D_0^2}\right)^2$$

Setting $z = h_{max}$ and $\dot m_{out} = \dot m_{in}$ (the incoming flow rate) gives the desired relation for the maximum height the water will reach in the tank,

$$h_{max}=\frac{1}{2g}\left(\frac{4\dot m_{in}}{\rho\pi D_0^2}\right)^2$$

(*b*) The amount of water that flows through the orifice and the increase in the amount of water in the tank during a differential time interval *dt* are

$$dm_{out}=\dot m_{out}dt=\rho\frac{\pi D_0^2}{4}\sqrt{2gz}\,dt$$

$$dm_{tank}=\rho A_{tank}dz=\rho\frac{\pi D_T^2}{4}dz$$

The amount of water that enters the tank during *dt* is $dm_{in}=\dot m_{in}dt$ (Recall that $\dot m_{in}$ = constant). Substituting them into the conservation of mass relation $dm_{tank}=dm_{in}-dm_{out}$ gives

$$dm_{tank}=\dot m_{in}dt-\dot m_{out}dt \quad\rightarrow\quad \rho\frac{\pi D_T^2}{4}dz=\left(\dot m_{in}-\rho\frac{\pi D_0^2}{4}\sqrt{2gz}\right)dt$$

Separating the variables, and integrating it from $z = 0$ at $t = 0$ to $z = z$ at time $t = t$ gives

$$\frac{\frac{1}{4}\rho\pi D_T^2dz}{\dot m_{in}-\frac{1}{4}\rho\pi D_0^2\sqrt{2gz}}=dt \quad\rightarrow\quad \int_{z=0}^{z}\frac{\frac{1}{4}\rho\pi D_T^2dz}{\dot m_{in}-\frac{1}{4}\rho\pi D_0^2\sqrt{2gz}}=\int_{t=0}^{t}dt=t$$

Performing the integration, the desired relation between the water height *z* and time *t* is obtained to be

$$\boxed{\frac{\frac{1}{2}\rho\pi D_T^2}{(\frac{1}{4}\rho\pi D_0^2\sqrt{2g})^2}\left(\frac{1}{4}\rho\pi D_0^2\sqrt{2gz}-\dot m_{in}\ln\frac{\dot m_{in}-\frac{1}{4}\rho\pi D_0^2\sqrt{2gz}}{\dot m_{in}}\right)=t}$$

Discussion Note that this relation is implicit in *z*, and thus we can't obtain a relation in the form $z = f(t)$. Substituting a *z* value in the left side gives the time it takes for the fluid level in the tank to reach that level. Equation solvers such as EES can easily solve implicit equations like this.

5-50E

Solution Water flows through a horizontal pipe that consists of two sections at a specified rate. The differential height of a mercury manometer placed between the two pipe sections is to be determined.

Assumptions **1**The flow through the pipe is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The losses in the reducing section are negligible.

Properties The densities of mercury and water are ρ_{Hg} = 847 lbm/ft^3 and ρ_w = 62.4 lbm/ft^3.

Analysis We take points 1 and 2 along the centerline of the pipe over the two tubes of the manometer. Noting that $z_1 = z_2$, the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_1-P_2=\frac{\rho_w(V_2^2-V_1^2)}{2} \qquad (1)$$

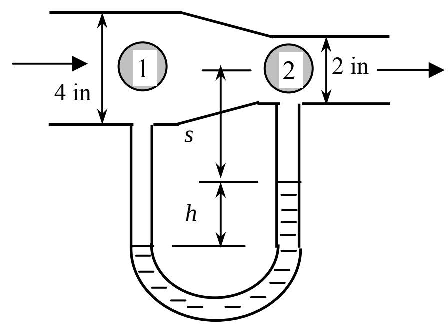
We let the differential height of the mercury manometer be h and the distance between the centerline and the mercury level in the tube where mercury is raised be s. Then the pressure difference $P_2 - P_1$ can also be expressed as

$$P_1+\rho_w g(s+h)=P_2+\rho_w gs+\rho_{Hg}gh \quad\rightarrow\quad P_1-P_2=(\rho_{Hg}-\rho_w)gh \qquad (2)$$

Combining Eqs. (1) and (2) and solving for h,

$$\frac{\rho_w(V_2^2-V_1^2)}{2}=(\rho_{Hg}-\rho_w)gh \quad\rightarrow\quad h=\frac{\rho_w(V_2^2-V_1^2)}{2g(\rho_{Hg}-\rho_w)}=\frac{V_2^2-V_1^2}{2g(\rho_{Hg}/\rho_w-1)}$$

Calculating the velocities and substituting,



$$V_1=\frac{\dot V}{A_1}=\frac{\dot V}{\pi D_1^2/4}=\frac{2.4\text{ gal/s}}{\pi(4/12\text{ ft})^2/4}\left(\frac{0.13368\text{ ft}^3}{1\text{ gal}}\right)=3.676\text{ ft/s}$$

$$V_2=\frac{\dot V}{A_2}=\frac{\dot V}{\pi D_2^2/4}=\frac{2.4\text{ gal/s}}{\pi(2/12\text{ ft})^2/4}\left(\frac{0.13368\text{ ft}^3}{1\text{ gal}}\right)=14.71\text{ ft/s}$$

$$h=\frac{(14.71\text{ ft/s})^2-(3.676\text{ ft/s})^2}{2(32.2\text{ ft/s}^2)(847/62.4-1)}=0.2504\text{ ft}=\mathbf{3.0\text{ in}}$$

Therefore, the differential height of the mercury column will be 3.0 in.

Discussion In reality, there are frictional losses in the pipe, and the pressure at location 2 will actually be smaller than that estimated here, and therefore h will be larger than that calculated here.

5-51

Solution An airplane is flying at a certain altitude at a given speed. The pressure on the stagnation point on the nose of the plane is to be determined, and the approach to be used at high velocities is to be discussed.

Assumptions **1** The air flow over the aircraft is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Standard atmospheric conditions exist. **3** The wind effects are negligible.

Properties The density of the atmospheric air at an elevation of 12,000 m is $\rho = 0.312$ kg/m^3.

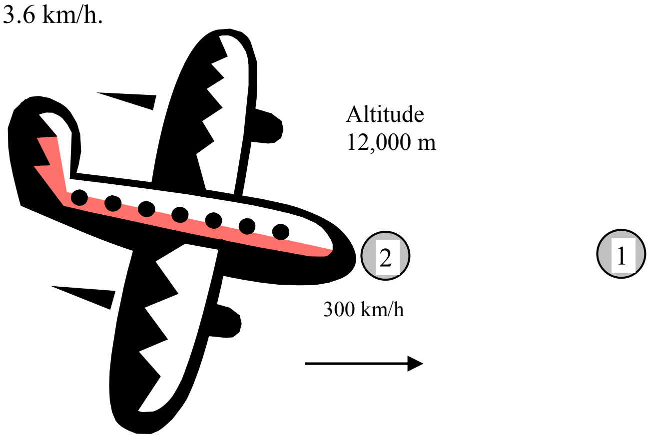
Analysis We take point 1 well ahead of the plane at the level of the nose, and point 2 at the nose where the flow comes to a stop. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad \frac{V_1^2}{2g} = \frac{P_2 - P_1}{\rho g} \quad \rightarrow \quad \frac{V_1^2}{2} = \frac{P_{stag} - P_{\text{atm}}}{\rho} = \frac{P_{\text{stag, gage}}}{\rho}$$

Solving for $P_{\text{stag, gage}}$ and substituting,

$$P_{\text{stag, gage}} = \frac{\rho V_1^2}{2} = \frac{(0.312\ \text{kg/m}^3)(300/3.6\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right) = 1083\ \text{N/m}^2 = \mathbf{1083\ Pa}$$

since 1 Pa = 1 N/m^2 and 1 m/s = 3.6 km/h.



Discussion A flight velocity of 1050 km/h = 292 m/s corresponds to a Mach number much greater than 0.3 (the speed of sound is about 340 m/s at room conditions, and lower at higher altitudes, and thus a Mach number of 292/340 = 0.86). Therefore, the flow can no longer be assumed to be incompressible, and the Bernoulli equation given above cannot be used. This problem can be solved using the modified Bernoulli equation that accounts for the effects of compressibility, assuming isentropic flow.

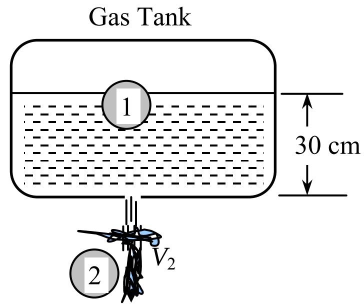
5-52

Solution The bottom of a car hits a sharp rock and a small hole develops at the bottom of its gas tank. For a given height of gasoline, the initial velocity of the gasoline out of the hole is to be determined. Also, the variation of velocity with time and the effect of the tightness of the lid on flow rate are to be discussed.

Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The air space in the tank is at atmospheric pressure. **3** The splashing of the gasoline in the tank during travel is not considered.

Analysis This problem involves the conversion of flow, kinetic, and potential energies to each other without involving any pumps, turbines, and wasteful components with large frictional losses, and thus it is suitable for the use of the Bernoulli equation. We take point 1 to be at the free surface of gasoline in the tank so that $P_1 = P_{atm}$ (open to the atmosphere) $V_1 \cong 0$ (the tank is large relative to the outlet), and $z_1 = 0.3$ m and $z_2 = 0$ (we take the reference level at the hole. Also, $P_2 = P_{atm}$ (gasoline discharges into the atmosphere). Then the Bernoulli equation simplifies to

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



Solving for V_2 and substituting,

$$V_2 = \sqrt{2gz_1} = \sqrt{2(9.81\ \text{m/s}^2)(0.3\ \text{m})} = \mathbf{2.43\ m/s}$$

Therefore, the gasoline will initially leave the tank with a velocity of 2.43 m/s.

Discussion The Bernoulli equation applies along a streamline, and streamlines generally do not make sharp turns. The velocity will be less than 2.43 m/s since the hole is probably sharp-edged and it will cause some head loss. As the gasoline level is reduced, the velocity will decrease since velocity is proportional to the square root of liquid height. If the lid is tightly closed and no air can replace the lost gasoline volume, the pressure above the gasoline level will be reduced, and the velocity will be decreased.

5-53

Solution The water in an above the ground swimming pool is to be emptied by unplugging the orifice of a horizontal pipe attached to the bottom of the pool. The maximum discharge rate of water is to be determined.

Assumptions **1** The orifice has a smooth entrance, and all frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

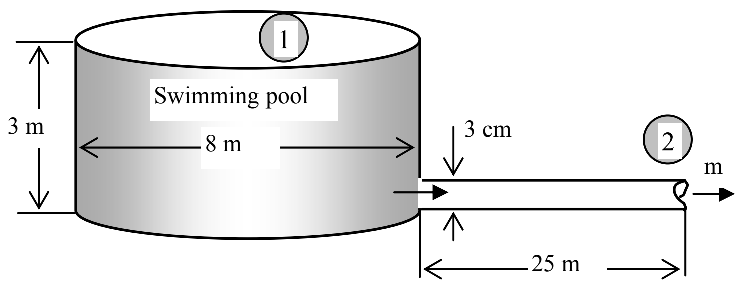
Analysis We take point 1 at the free surface of the pool, and point 2 at the exit of pipe. We take the reference level at the pipe exit (z_2 = 0). Noting that the fluid at both points is open to the atmosphere (and thus $P_1 = P_2 = P_{atm}$) and that the fluid velocity at the free surface is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

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

The maximum discharge rate occurs when the water height in the pool is a maximum, which is the case at the beginning and thus $z_1 = h$. Substituting, the maximum flow velocity and discharge rate become

$$V_{2,\max}=\sqrt{2gh}=\sqrt{2(9.81\text{ m/s}^2)(3\text{ m})}=7.67\text{ m/s}$$

$$\dot{V}_{\max}=A_{\text{pipe}}V_{2,\max}=\frac{\pi D^2}{4}V_{2,\max}=\frac{\pi(0.03\text{ m})^2}{4}(7.67\text{ m/s})=0.00542\text{ m}^3/\text{s}=\mathbf{5.42\ L/s}$$



Discussion The result above is obtained by disregarding all frictional effects. The actual flow rate will be less because of frictional effects during flow and the resulting pressure drop. Also, the flow rate will gradually decrease as the water level in the pipe decreases.

5-54

Solution The water in an above the ground swimming pool is to be emptied by unplugging the orifice of a horizontal pipe attached to the bottom of the pool. The time it will take to empty the tank is to be determined.

Assumptions **1** The orifice has a smooth entrance, and all frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Analysis We take point 1 at the free surface of water in the pool, and point 2 at the exit of pipe. We take the reference level at the pipe exit ($z_2 = 0$). 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 very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

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

For generality, we express the water height in the pool at any time t by z, and the discharge velocity by $V_2=\sqrt{2gz}$. Note that water surface in the pool moves down as the pool drains, and thus z is a variable whose value changes from h at the beginning to *0* when the pool is emptied completely.

We denote the diameter of the orifice by D, and the diameter of the pool by D_o. The flow rate of water from the pool is obtained by multiplying the discharge velocity by the orifice cross-sectional area,

$$\dot{V}=A_{\text{orifice}}V_2=\frac{\pi D^2}{4}\sqrt{2gz}$$

Then the amount of water that flows through the orifice during a differential time interval dt is

$$dV=\dot{V}dt=\frac{\pi D^2}{4}\sqrt{2gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the pool,

$$dV=A_{\text{tank}}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the pool during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

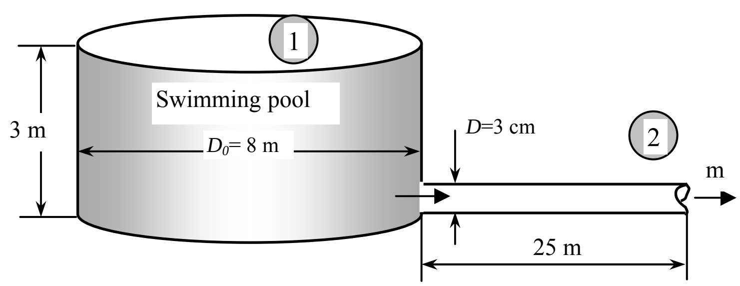
$$\frac{\pi D^2}{4}\sqrt{2gz}\,dt=-\frac{\pi D_0^2}{4}dz \quad\rightarrow\quad dt=-\frac{D_0^2}{D^2}\sqrt{\frac{1}{2gz}}dz=-\frac{D_0^2}{D^2\sqrt{2g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = h$ to $t = t_f$ when $z = 0$ (completely drained pool) gives

$$\int_{t=0}^{t_f}dt=-\frac{D_0^2}{D^2\sqrt{2g}}\int_{z=z_1}^{0}z^{-1/2}dz \quad\rightarrow\quad t_f=-\frac{D_0^2}{D^2\sqrt{2g}}\left|\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0}=\frac{2D_0^2}{D^2\sqrt{2g}}\sqrt{h}=\frac{D_0^2}{D^2}\sqrt{\frac{2h}{g}}$$

Substituting, the draining time of the pool will be

$$t_f=\frac{(8\text{ m})^2}{(0.03\text{ m})^2}\sqrt{\frac{2(3\text{ m})}{9.81\text{ m/s}^2}}=55{,}600\text{ s}=\mathbf{15.4\ h}$$



Discussion This is the minimum discharging time since it is obtained by neglecting all friction; the actual discharging time will be longer. Note that the discharging time is inversely proportional to the square of the orifice diameter. Therefore, the discharging time can be reduced to one-fourth by doubling the diameter of the orifice.

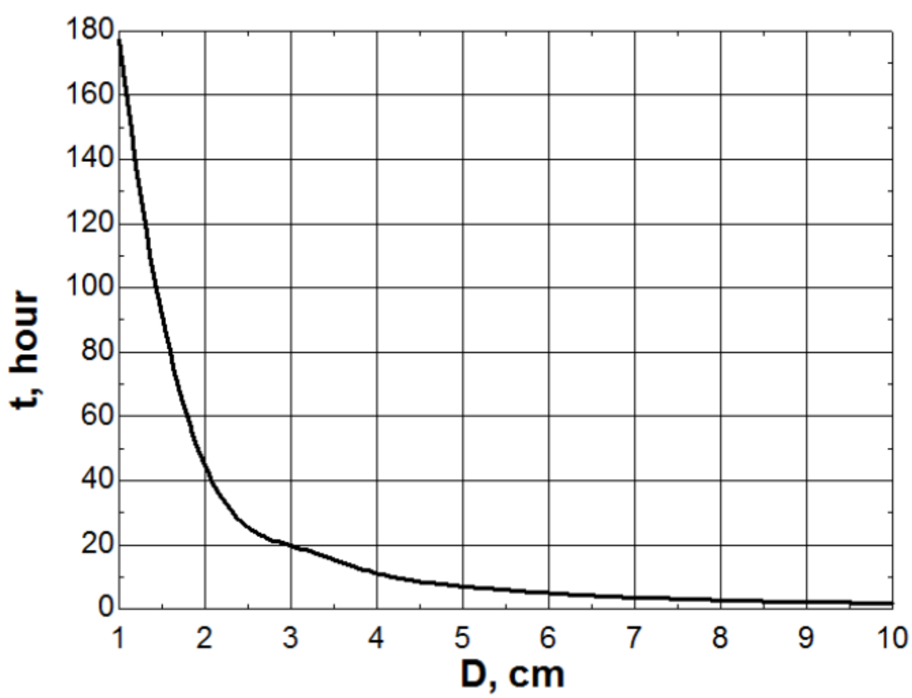
5-55

Solution The previous problem is reconsidered. The effect of the discharge pipe diameter on the time required to empty the pool completely as the diameter varies from 1 to 10 cm in increments of 1 cm is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
h=2 "m"
D=d_pipe/100 "m"
D_pool=10 "m"
V_initial=SQRT(2*g*h) "m/s"
Ac=pi*D^2/4
V_dot=Ac*V_initial*1000 "m3/s"
t=(D_pool/D)^2*SQRT(2*h/g)/3600 "hour"
```

Pipe diameter *D*, m	Discharge time *t*, h
1	177.4
2	44.3
3	19.7
4	11.1
5	7.1
6	4.9
7	3.6
8	2.8
9	2.2
10	1.8



Discussion As you can see from the plot, the discharge time is drastically reduced by increasing the pipe diameter.

5-56

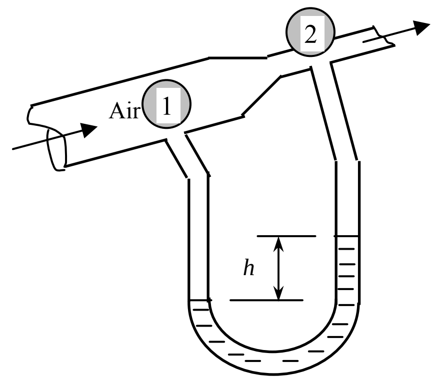
Solution Air flows upward at a specified rate through an inclined pipe whose diameter is reduced through a reducer. The differential height between fluid levels of the two arms of a water manometer attached across the reducer is to be determined.

Assumptions **1** The flow through the duct is steady, incompressible and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** Air is an ideal gas. **3** The effect of air column on the pressure change is negligible because of its low density. **4** The air flow is parallel to the entrance of each arm of the manometer, and thus no dynamic effects are involved.

Properties We take the density of water to be $\rho = 1000$ kg/m^3. The gas constant of air is $R = 0.287$ kPa·m^3/kg·K.

Analysis We take points 1 and 2 at the lower and upper connection points, respectively, of the two arms of the manometer, and take the lower connection point as the reference level. Noting that the effect of elevation on the pressure change of a gas is negligible, the application of the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\to\quad P_1-P_2=\rho_{\text{air}}\frac{V_2^2-V_1^2}{2}$$



where $\rho_{air}=\frac{P}{RT}=\frac{105\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(37+273\text{ K})}=1.180\text{ kg/m}^3$

$$V_1=\frac{\dot V}{A_1}=\frac{\dot V}{\pi D_1^2/4}=\frac{0.065\text{ m}^3/\text{s}}{\pi(0.06\text{ m})^2/4}=22.99\text{ m/s}$$

$$V_2=\frac{\dot V}{A_2}=\frac{\dot V}{\pi D_2^2/4}=\frac{0.065\text{ m}^3/\text{s}}{\pi(0.04\text{ m})^2/4}=51.73\text{ m/s}$$

Substituting,

$$P_1-P_2=(1.180\text{ kg/m}^3)\frac{(51.73\text{ m/s})^2-(22.99\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)=1267\text{ N/m}^2=1267\text{ Pa}$$

The differential height of water in the manometer corresponding to this pressure change is determined from $\Delta P=\rho_{water}gh$ to be

$$h=\frac{P_1-P_2}{\rho_{water}g}=\frac{1267\text{ N/m}^2}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)=0.1291\text{ m}=\mathbf{12.9\ cm}$$

Discussion When the effect of air column on pressure change is considered, the pressure change becomes

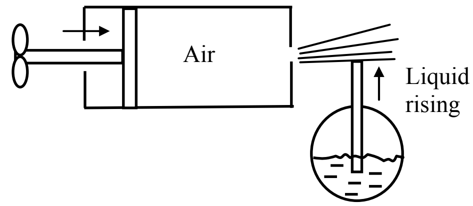
$$\begin{aligned}P_1-P_2&=\frac{\rho_{air}(V_2^2-V_1^2)}{2}+\rho_{air}g(z_2-z_1)\\&=(1.180\text{ kg/m}^3)\left[\frac{(51.73\text{ m/s})^2-(22.99\text{ m/s})^2}{2}+(9.81\text{ m/s}^2)(0.2\text{ m})\right]\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\\&=(1267+2)\text{ N/m}^2=1269\text{ N/m}^2=1269\text{ Pa}\end{aligned}$$

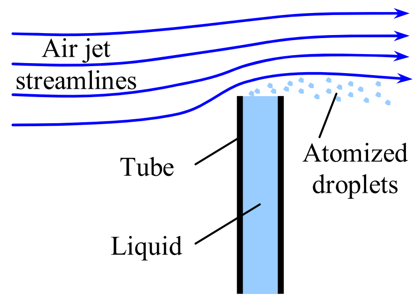
This difference between the two results (1267 and 1269 Pa) is less than 1%. Therefore, the effect of air column on pressure change is, indeed, negligible as assumed. In other words, the pressure change of air in the duct is almost entirely due to velocity change, and the effect of elevation change is negligible. Also, if we were to account for the Δz of air flow, then it would be more proper to account for the Δz of air in the manometer by using ρ_{water} - ρ_{air} instead of ρ_{water} when calculating h. The additional air column in the manometer tends to cancel out the change in pressure due to the elevation difference in the flow in this case.

5-57

Solution A hand-held bicycle pump with a liquid reservoir is used as an atomizer by forcing air at a high velocity through a small hole. We are to explain how the liquid gets sucked up the tube.

Assumptions **1** The flows of air and water are steady and nearly incompressible. **2** Air is an ideal gas. **3** The liquid reservoir is open to the atmosphere. **4** The device is held horizontally. **5** The water velocity through the tube is low (the water in the tube is nearly hydrostatic).





Analysis At first glance, we are tempted to use the Bernoulli equation, thinking that the pressure in the air jet would be lower than atmospheric due to its high speed. However, as stated in the problem statement, the pressure through an incompressible jet exposed to the atmosphere is nearly atmospheric pressure everywhere. Thus, in the absence of the tube, the pressure in the air jet just above the tube would be nearly atmospheric. Meanwhile, the pressure at the liquid surface is also atmospheric. Applying hydrostatics from the liquid surface to the top of the tube reveals that the pressure at the top of the tube must be lower than atmospheric pressure by more than ρgh in order to suck the liquid up the tube. So, what causes the liquid to rise? It turns out that the answer has to do with **streamline curvature**. As the close-up sketch illustrates, the **air streamlines must curve around the top of the tube**. Since pressure decreases towards the center of curvature in a flow with curved streamlines, the pressure at the top of the tube must be less than atmospheric. At high enough air jet speed, the pressure is low enough not only to suck the liquid to the top of the tube, but also to break up the liquid at the top of the tube into small droplets, thereby "atomizing" the liquid into a spray of liquid droplets.

Discussion If the geometry of the top of the tube were known, we could approximate the flow as irrotational and apply the techniques of potential flow analysis (Chap. 10) to estimate the pressure at the top of the tube.

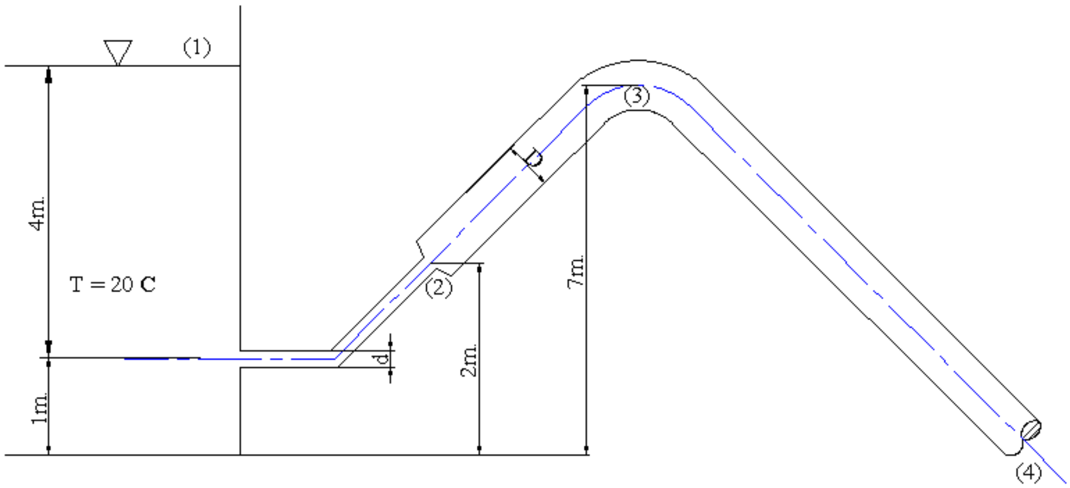
5-58

Solution Water is siphoned from a reservoir. The minimum flow rate that can be achieved without cavitation occurring in the piping system and the maximum elevation of the highest point of the piping system to avoid cavitation are to be determined.

Assumptions **1** The flow through the pipe is steady, incompressible and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

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

Analysis



For T = 20 ^{0}C , P_V = 2.338 x 10^3 Pa (abs)

d = 10 cm. , D = 16 cm.

Applying Bernoulli Eq. between (1) and (4)

$$\frac{P_1}{\gamma}+Z_1+\frac{V_1^2}{2g}=\frac{P_4}{\gamma}+Z_4+\frac{V_4^2}{2g} \quad , \quad V_4=\sqrt{2gh_1}=\sqrt{2g(1+4)}$$

V_4 = 9.904 m/s

On the other hand, from the continuity,

$$A_dV_d=A_DV_D \quad , \quad V_d=(\frac{A_D}{A_d})V_D=\frac{D^2}{d^2}V_D$$

$$V_d=(\frac{16}{10})^2V_D$$

$$V_d=2.56V_D=2.56x9.904=25.35 \ m/s$$

We should check if these velocities would be possible,

Bernoulli Eq. from (1) to (2) yields

$$\frac{P_{1m}}{\gamma}+Z_1+\frac{V_1^2}{2g}=\frac{P_{2m}}{\gamma}+Z_2+\frac{V_2^2}{2g}$$

$$\frac{101325}{9810}+5+0=\frac{P_{2m}}{\gamma}+2+\frac{23.35^2}{19.62}$$

$$15.328=\frac{P_{2m}}{\gamma}+29.789 \quad , \quad \frac{P_{2m}}{\gamma}=-14.461m.$$

Since $\frac{P_{2m}}{\gamma} < 0$, the velocity V_d cannot be 23.35 m/s. Applying Bernoulli Eq. from (1) to(2)

$$\frac{101325}{9810} + 5 = 2 + \frac{2.338.10^3}{9810} + \frac{V_{d\max}^2}{2g}$$

$$\frac{V_{d\max}^2}{2g} = 13.09$$

$V_{max} \approx 16$ m/s. Therefore the velocity will never exceed 16 m/s. Accordingly;

$$\dot{V} = A_d V_d = \pi \frac{0.1^2}{4} 16 = \mathbf{0.125\ m^3/s}$$

(b) For a maximum Z_3 , the absolute pressure $P_{3\min} = 2338\ Pa(abs)$

$$\frac{P_{1m}}{\gamma} + Z_1 + \frac{V_1^2}{2g} = constant$$, therefore

$\frac{P_3}{\gamma}$ and $\frac{V_3^2}{2g}$ must be minimum.

$$\frac{P_{3\min}}{\gamma} = \frac{2338}{9810} = 0.238m.$$

Applying Bernoulli Eq. from (1) to (3)

$$\frac{101325}{9810} + 5 + 0 = Z_{3.\max} + 0.238 + \frac{V_D^2}{2g}$$

From the first part,

$$V_D = \frac{d^2}{D^2} V_d = (\frac{10}{16})^2 16$$

$V_D = 6.25$ m/s

Therefore $Z_{3.\max} = \frac{101325}{9810} + 5 - 0.238 - \frac{6.25^2}{19.62}$

$Z_{3.\max} \approx$ **13 m**

5-59

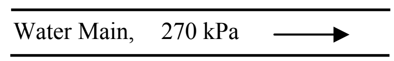
Solution The gage pressure in the water mains of a city at a particular location is given. It is to be determined if this main can serve water to neighborhoods that are at a given elevation relative to this location.

Assumptions Water is incompressible and thus its density is constant.

Properties We take the density of water to be $\rho = 1000\ \text{kg/m}^3$.

Analysis Noting that the gage pressure at a dept of h in a fluid is given by $P_{gage} = \rho_{water} gh$, the height of a fluid column corresponding to a gage pressure of 270 kPa is determined to be

$$h = \frac{P_{gage}}{\rho_{water} g} = \frac{270{,}000\ \text{N/m}^2}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) = 27.5\ \text{m}$$

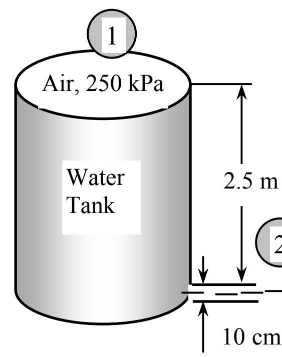


which is higher than 25 m. Therefore, this main **can** serve water to neighborhoods that are 25 m above this location.

Discussion Note that h must be much greater than 25 m for water to have enough pressure to serve the water needs of the neighborhood.

5-60

Solution Water discharges to the atmosphere from the orifice at the bottom of a pressurized tank. Assuming frictionless flow, the discharge rate of water from the tank is to be determined.



Assumptions **1** The orifice has a smooth entrance, and thus the frictional losses are negligible. **2** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Properties We take the density of water to be $1000\ \text{kg/m}^3$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of orifice, which is also taken to be the reference level ($z_2 = 0$). Noting that the fluid velocity at the free surface is very low ($V_1 \cong 0$) and water discharges into the atmosphere (and thus $P_2 = P_{atm}$), the Bernoulli equation simplifies to

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

Solving for V_2 and substituting, the discharge velocity is determined to

$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho} + 2gz_1} = \sqrt{\frac{2(250-100)\ \text{kPa}}{1000\ \text{kg/m}^3}\left(\frac{1000\ \text{N/m}^2}{1\ \text{kPa}}\right)\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right) + 2(9.81\ \text{m/s}^2)(2.5\ \text{m})} = 18.7\ \text{m/s}$$

Then the initial rate of discharge of water becomes

$$\dot{V} = A_{orifice} V_2 = \frac{\pi D^2}{4} V_2 = \frac{\pi (0.10\ \text{m})^2}{4}(18.7\ \text{m/s}) = \mathbf{0.147\ m^3/s}$$

Discussion Note that this is the maximum flow rate since the frictional effects are ignored. Also, the velocity and the flow rate will decrease as the water level in the tank decreases.

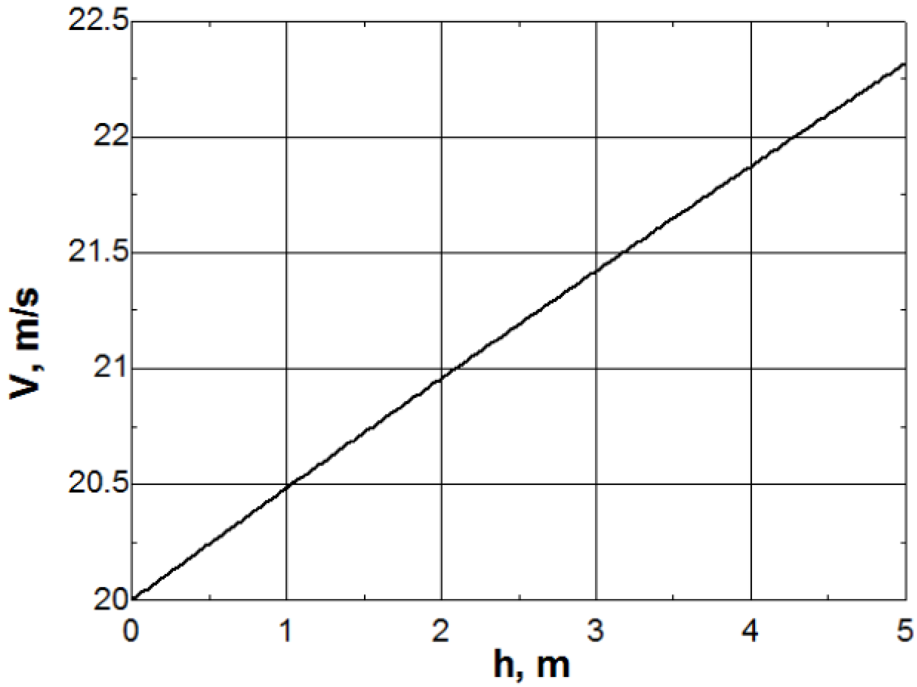
5-61

Solution The previous problem is reconsidered. The effect of water height in the tank on the discharge velocity as the water height varies from 0 to 5 m in increments of 0.5 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
d=0.10 "m"
P1=300 "kPa"
P_atm=100 "kPa"
V=SQRT(2*(P1-P_atm)*1000/rho+2*g*h)
Ac=pi*D^2/4
V_dot=Ac*V
```

h, m	*V*, m/s	$\dot{V}$, m^3/s
0.00	20.0	0.157
0.50	20.2	0.159
1.00	20.5	0.161
1.50	20.7	0.163
2.00	21.0	0.165
2.50	21.2	0.166
3.00	21.4	0.168
3.50	21.6	0.170
4.00	21.9	0.172
4.50	22.1	0.174
5.00	22.3	0.175



Discussion Velocity appears to change nearly linearly with *h* in this range of data, but the relationship is *not* linear.

5-62E

Solution Air is flowing through a venturi meter with known diameters and measured pressures. A relation for the flow rate is to be obtained, and its numerical value is to be determined.

Assumptions **1**The flow through the venturi is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The effect of air column on the pressure change is negligible because of its low density, and thus the pressure can be assumed to be uniform at a given cross-section of the venturi meter (independent of elevation change). **3** The flow is horizontal (this assumption is usually unnecessary for gas flow.).

Properties The density of air is given to be $\rho = 0.075$ lbm/ft^3.

Analysis We take point 1 at the main flow section and point 2 at the throat along the centerline of the venturi meter. Noting that $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

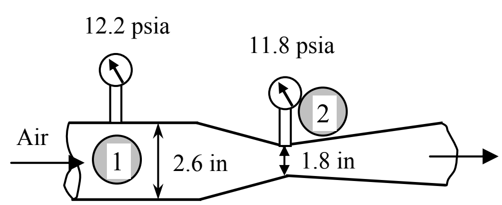
$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad P_1-P_2=\rho\frac{V_2^2-V_1^2}{2} \qquad (1)$$

The flow is assumed to be incompressible and thus the density is constant. Then the conservation of mass relation for this single stream steady flow device can be expressed as

$$\dot V_1=\dot V_2=\dot V \quad\rightarrow\quad A_1V_1=A_2V_2=\dot V \quad\rightarrow\quad V_1=\frac{\dot V}{A_1} \quad \text{and} \quad V_2=\frac{\dot V}{A_2} \qquad (2)$$

Substituting into Eq. (1),

$$P_1-P_2=\rho\frac{(\dot V/A_2)^2-(\dot V/A_1)^2}{2}=\frac{\rho\dot V^2}{2A_2^2}\left(1-\frac{A_2^2}{A_1^2}\right)$$



Solving for $\dot V$ gives the desired relation for the flow rate,

$$\dot V=A_2\sqrt{\frac{2(P_1-P_2)}{\rho[1-(A_2/A_1)^2]}} \qquad (3)$$

The flow rate for the given case can be determined by substituting the given values into this relation to be

$$\dot V=\frac{\pi D_2^2}{4}\sqrt{\frac{2(P_1-P_2)}{\rho[1-(D_2/D_1)^4]}}=\frac{\pi(1.8/12\text{ ft})^2}{4}\sqrt{\frac{2(12.2-11.8)\text{ psi}}{(0.075\text{ lbm/ft}^3)[1-(1.8/2.6)^4]}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)}$$

$$= \mathbf{4.48\ ft^3/s}$$

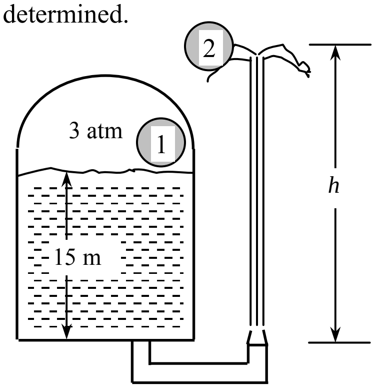
Discussion Venturi meters are commonly used as flow meters to measure the flow rate of gases and liquids by simply measuring the pressure difference P_1 - P_2 by a manometer or pressure transducers. The actual flow rate will be less than the value obtained from Eq. (3) because of the friction losses along the wall surfaces in actual flow. But this difference can be as little as 1% in a well-designed venturi meter. The effects of deviation from the idealized Bernoulli flow can be accounted for by expressing Eq. (3) as

$$\dot V=C_cA_2\sqrt{\frac{2(P_1-P_2)}{\rho[1-(A_2/A_1)^2]}}$$

where C_c is the *venturi discharge coefficient* whose value is less than 1 (it is as large as 0.99 for well-designed venturi meters in certain ranges of flow). For Re > 10^5, the value of venturi discharge coefficient is usually greater than 0.96.

5-63

Solution The water height in an airtight pressurized tank is given. A hose pointing straight up is connected to the bottom of the tank. The maximum height to which the water stream could rise is to be determined.



Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The friction between the water and air is negligible.

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

Analysis We take point 1 at the free surface of water in the tank, and point 2 at the top of the water trajectory. Also, we take the reference level at the bottom of the tank. At the top of the water trajectory $\mathbf{V}_2 = 0$, and atmospheric pressure pertains. Noting that $z_1 = 20$ m, $P_{1,\text{gage}} = 2$ atm, $P_2 = P_{\text{atm}}$, and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the Bernoulli equation between these two points simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad \rightarrow \quad \frac{P_1}{\rho g}+z_1=\frac{P_{atm}}{\rho g}+z_2 \quad \rightarrow \quad z_2=\frac{P_1-P_{atm}}{\rho g}+z_1=\frac{P_{1,\text{gage}}}{\rho g}+z_1$$

Substituting,

$$z_2=\frac{3\text{ atm}}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{101{,}325\text{ N/m}^2}{1\text{ atm}}\right)\left(\frac{1\text{ kg}\cdot\text{m/s}^2}{1\text{ N}}\right)+15=\mathbf{46.0\text{ m}}$$

Therefore, the water jet can rise as high as 46.0 m into the sky from the ground.

Discussion The result obtained by the Bernoulli equation represents the upper limit, and should be interpreted accordingly. It tells us that the water cannot possibly rise more than 46.0 m (giving us an upper limit), and in all likelihood, the rise will be much less because of frictional losses.

5-64

Solution A Pitot-static probe equipped with a water manometer is held parallel to air flow, and the differential height of the water column is measured. The flow velocity of air is to be determined.

Assumptions **1**The flow of air is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The effect of air column on the pressure change is negligible because of its low density, and thus the air column in the manometer can be ignored.

Properties We take the density of water to be ρ = 1000 kg/m^3. The density of air is given to be 1.16 kg/m^3.

Analysis We take point 1 on the side of the probe where the entrance is parallel to flow and is connected to the static arm of the Pitot-static probe, and point 2 at the tip of the probe where the entrance is normal to flow and is connected to the dynamic arm of the Pitot-static probe. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

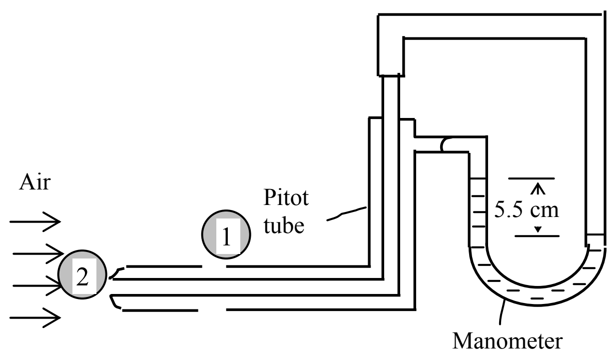
$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+\frac{V_1^2}{2g}=\frac{P_2}{\rho g} \quad\rightarrow\quad V_1=\sqrt{\frac{2(P_2-P_1)}{\rho_{air}}} \qquad (1)$$

The pressure rise at the tip of the Pitot-static probe is simply the pressure change indicated by the differential water column of the manometer,

$$P_2-P_1=\rho_{\text{water}}gh \qquad (2)$$

Combining Eqs. (1) and (2) and substituting, the flow velocity is determined to be

$$V_1=\sqrt{\frac{2\rho_{\text{water}}gh}{\rho_{\text{air}}}}=\sqrt{\frac{2(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(0.055\ \text{m})}{1.16\ \text{kg/m}^3}}=\mathbf{30.5\ m/s}$$



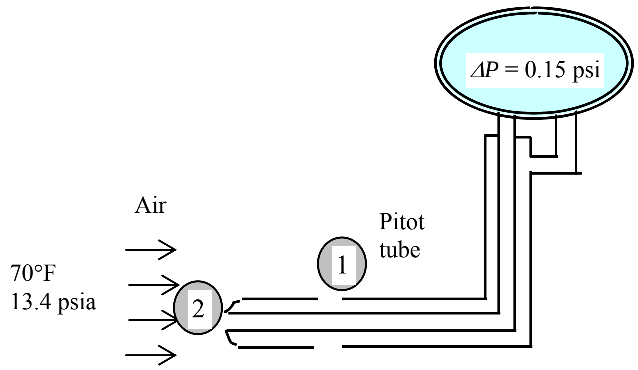
Discussion Note that flow velocity in a pipe or duct can be measured easily by a Pitot-static probe by inserting the probe into the pipe or duct parallel to flow, and reading the differential height. Also note that this is the velocity at the location of the tube. Several readings at several locations in a cross-section may be required to determine the mean flow velocity.

5-65E

Solution A Pitot-static probe equipped with a differential pressure gage is used to measure the air velocity in a duct. For a given differential pressure reading, the flow velocity of air is to be determined.

Assumptions The flow of air is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable).

Properties The gas constant of air is R = 0.3704 psia·ft^3/lbm·R.



Analysis We take point 1 on the side of the probe where the entrance is parallel to flow and is connected to the static arm of the Pitot-static probe, and point 2 at the tip of the probe where the entrance is normal to flow and is connected to the dynamic arm of the Pitot-static probe. Noting that point 2 is a stagnation point and thus $V_2 = 0$ and $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

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

where

$$\rho=\frac{P}{RT}=\frac{13.4\text{ psia}}{(0.3704\text{ psia}\cdot\text{ft}^3/\text{lbm}\cdot\text{R})(70+460\text{ R})}=0.0683\text{ lbm/ft}^3$$

Substituting the given values, the flow velocity is determined to be

$$V_1=\sqrt{\frac{2(0.15\text{ psi})}{0.0683\text{ lbm/ft}^3}\left(\frac{144\text{ lbf/ft}^2}{1\text{ psi}}\right)\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)}=\mathbf{143\ ft/s}$$

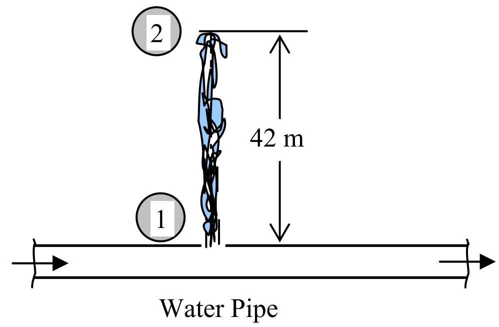
Discussion Note that flow velocity in a pipe or duct can be measured easily by a Pitot-static probe by inserting the probe into the pipe or duct parallel to flow, and reading the pressure differential. Also note that this is the velocity at the location of the tube. Several readings at several locations in a cross-section may be required to determine the mean flow velocity.

5-66

Solution A water pipe bursts as a result of freezing, and water shoots up into the air a certain height. The gage pressure of water in the pipe is to be determined.

Assumptions **1** The flow is steady, incompressible, and irrotational with negligible frictional effects (so that the Bernoulli equation is applicable). **2** The water pressure in the pipe at the burst section is equal to the water main pressure. **3** Friction between the water and air is negligible. **4** The irreversibilities that may occur at the burst section of the pipe due to abrupt expansion are negligible.

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



Analysis This problem involves the conversion of flow, kinetic, and potential energies to each other without involving any pumps, turbines, and wasteful components with large frictional losses, and thus it is suitable for the use of the Bernoulli equation. The water height will be maximum under the stated assumptions. The velocity inside the hose is relatively low ($V_1 \cong 0$) and we take the burst section of the pipe as the reference level ($z_1 = 0$). At the top of the water trajectory $V_2 = 0$, and atmospheric pressure pertains. Then the Bernoulli equation simplifies to

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}=\frac{P_{atm}}{\rho g}+z_2 \quad\rightarrow\quad \frac{P_1-P_{atm}}{\rho g}=z_2 \quad\rightarrow\quad \frac{P_{1,\text{gage}}}{\rho g}=z_2$$

Solving for $P_{1,\text{gage}}$ and substituting,

$$P_{1,\text{gage}}=\rho g z_2=(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(42\text{ m})\left(\frac{1\text{ kPa}}{1\text{ kN/m}^2}\right)\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)=\mathbf{412\text{ kPa}}$$

Therefore, the pressure in the main must be at least 412 kPa above the atmospheric pressure.

Discussion The result obtained by the Bernoulli equation represents a limit, since frictional losses are neglected, and should be interpreted accordingly. It tells us that the water pressure (gage) cannot possibly be less than 334 kPa (giving us a lower limit), and in all likelihood, the pressure will be much higher.

5-67

Solution A well-fitting piston with 4 small holes in a sealed water-filled cylinder is pushed to the right at a constant speed while the pressure in the right compartment remains constant. The force that needs to be applied to the piston to maintain this motion is to be determined.

Analysis When frictional effects are negligible, the pressures on both sides of the piston become identical:

$$F=P_{cyl,gage}A_p=P_{cyl,gage}\frac{\pi d_p^2}{4}=(50\text{ kPa})\frac{\pi(0.12\text{ m})^2}{4}=\mathbf{0.565\text{ kN}}$$

5-68

Solution We are to generate expressions for inlet pressure and throat pressure and velocity in a converging-diverging duct for the case where P_{outlet} and V_{inlet} are known and irreversibilities are ignored.

Assumptions **1** The flow is steady, incompressible, and two-dimensional. **2** We neglect irreversibilities such as friction. **3** **4** The duct is horizontal – elevation differences do not play a role in this analysis.

Analysis (*a*) We apply conservation of mass from the inlet to the outlet:

$$\rho V_{inlet} A_{inlet} = \rho V_{outlet} A_{outlet} \quad \rightarrow \quad V_{outlet} = V_{inlet}\frac{A_{inlet}}{A_{outlet}}$$

where density has dropped out because of the incompressible flow approximation. We do a similar analysis at the throat. Thus, the average inlet velocity and average throat velocity are

$$\boxed{V_{outlet} = V_{inlet}\frac{A_{inlet}}{A_{outlet}} \qquad V_{throat} = V_{inlet}\frac{A_{inlet}}{A_{throat}}}$$

To estimate the pressure at inlet, we neglect irreversibilities and apply the Bernoulli equation along a streamline from the inlet to the outlet,

$$P_{inlet} + \frac{1}{2}\rho V_{inlet}^2 + \cancel{\rho g z_{inlet}} = P_{outlet} + \frac{1}{2}\rho V_{outlet}^2 + \cancel{\rho g z_{outlet}} \quad \rightarrow \quad P_{inlet} = P_{outlet} + \frac{1}{2}\rho\left(V_{outlet}^2 - V_{inlet}^2\right)$$

and when we substitute the outlet velocity from conservation of mass we get

$$\boxed{P_{inlet} = P_{outlet} + \frac{1}{2}\rho\left(\left(V_{inlet}\frac{A_{inlet}}{A_{outlet}}\right)^2 - V_{inlet}^2\right) = \frac{1}{2}\rho V_{inlet}^2\left(\left(\frac{A_{inlet}}{A_{throat}}\right)^2 - 1\right)}$$

In like manner, we calculate the average pressure at any other axial location where the cross-sectional area is known. At the throat, for example,

$$P_{throat} = P_{outlet} + \frac{1}{2}\rho\left(V_{outlet}^2 - V_{throat}^2\right) = P_{outlet} + \frac{1}{2}\rho\left(\left(V_{inlet}\frac{A_{inlet}}{A_{outlet}}\right)^2 - \left(V_{inlet}\frac{A_{inlet}}{A_{throat}}\right)^2\right)$$

or, combining some terms,

$$\boxed{P_{throat} = P_{outlet} + \frac{1}{2}\rho\left(V_{inlet}A_{inlet}\right)^2\left(\left(\frac{1}{A_{outlet}}\right)^2 - \left(\frac{1}{A_{throat}}\right)^2\right)}$$

(*b*) In this analysis, we have not accounted for any irreversibilities, such as friction, but in any real flow, friction would lead to higher pressure drop in the duct and thus **the inlet pressure would have to be higher than predicted in order to overcome the additional losses due to friction**.

Discussion We must keep in mind that the Bernoulli equation is only an *approximation*. In Chap. 8 we learn how to approximate the additional pressure drop due to friction along the walls of a duct or pipe.

Energy Equation

5-69C

Solution We are to define and discuss useful pump head.

Analysis *Useful pump head* is the **useful power input to the pump expressed as an equivalent column height of fluid**. It is related to the useful pumping power input by $\boxed{h_{\text{pump}} = \frac{w_{\text{pump, u}}}{g} = \frac{\dot{W}_{\text{pump, u}}}{\dot{m}g}}$.

Discussion Part of the power supplied to the pump is *not* useful, but rather is wasted because of irreversible losses in the pump. This is the reason that pumps have a pump efficiency that is always less than one.

5-70C

Solution We are to analyze whether temperature can decrease during steady adiabatic flow of an incompressible fluid.

Analysis **It is *impossible* for the fluid temperature to decrease** during steady, incompressible, adiabatic flow of an incompressible fluid, since this would require the entropy of an adiabatic system to decrease, which would be a violation of the 2^{nd} law of thermodynamics.

Discussion The entropy of a fluid *can* decrease, but only if we remove heat.

5-71C

Solution We are to define and discuss irreversible head loss.

Analysis *Irreversible head loss* is the **loss of mechanical energy due to irreversible processes** (such as friction) **in piping expressed as an equivalent column height of fluid**, i.e., head. Irreversible head loss is related to the mechanical energy loss in piping by $\boxed{h_L = \frac{e_{\text{mech loss, piping}}}{g} = \frac{\dot{E}_{\text{mech loss, piping}}}{\dot{m}g}}$.

Discussion h_L is always positive. It can never be negative, since this would violate the second law of thermodynamics.

5-72C

Solution We are to determine if frictional effects are negligible in the steady adiabatic flow of an incompressible fluid if the temperature remains constant.

Analysis **Yes, the *frictional effects* are negligible** if the fluid temperature remains constant during steady, incompressible flow since any irreversibility such as friction would cause the entropy and thus temperature of the fluid to increase during adiabatic flow.

Discussion Thus, this scenario would never occur in real life since all fluid flows have frictional effects.

5-73C

Solution We are to define and discuss the kinetic energy correction factor.

Analysis The *kinetic energy correction factor* is a **correction factor to account for the fact that kinetic energy using average velocity is not the same as the actual kinetic energy using the actual velocity profile** (the square of a sum is not equal to the sum of the squares of its components). The effect of kinetic energy factor is usually negligible, especially for turbulent pipe flows. However, for laminar pipe flows, the effect of α is sometimes significant.

Discussion Even though the effect of ignoring α is usually insignificant, it is wise to keep α in our analyses to increase accuracy and so that we do not forget about it in situations where it *is* significant, such as in some laminar pipe flows.

5-74C

Solution We are to analyze the cause of some strange behavior of a water jet.

Analysis The problem does not state whether the water in the tank is open to the atmosphere or not. Let's assume that the water surface *is* exposed to atmospheric pressure. By the Bernoulli equation, the maximum theoretical height to which the water stream could rise is the tank water level, which is 20 meters above the ground. Since the water rises *above* the tank level, **the tank cover must be airtight, containing pressurized air above the water surface**. In other words, the water in the tank is *not* exposed to atmospheric pressure.

Discussion Alternatively, a pump would have to pressurize the water somewhere in the hose, but this is not allowed, based on the problem statement (only a hose is added).

5-75C

Solution We are to analyze a suggestion regarding a garden hose.

Analysis **Yes**. When water discharges from the hose at waist level, the head corresponding to the waist-knee vertical distance is wasted. When recovered, this elevation head is converted to velocity head, increasing the discharge velocity (and thus the flow rate) of water and thus reducing the filling time.

Discussion If you are still not convinced, imagine holding the hose outlet really high up. If the outlet elevation is greater than the upstream supply head, no water will flow at all. If you are concerned about head losses in the hose, yes, they will increase as the volume flow rate increases, but not enough to change our answer.

5-76C

Solution We are to analyze discharge of water from a tank under different conditions.

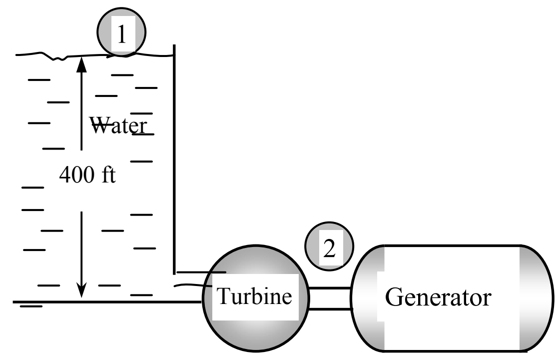
Analysis (*a*) **Yes**, the discharge velocity from the bottom valve will be higher since velocity is proportional to the square root of the vertical distance between the hole and the free surface. (*b*) **No**, the discharge rates of water will be the same since the total available head to drive the flow (elevation difference between the ground and the free surface of water in the tank) is the same for both cases.

Discussion Our answer to Part (b) does not change even if we consider head losses in the hose, because the hose is the same length in either case. Same hose, same length, same flow rate…yields the same head loss through the hose. *Note*: We are ignoring any effects of bends or curves in the hose – assume both cases have the same curves.

5-77E

Solution In a hydroelectric power plant, the elevation difference, the power generation, and the overall turbine-generator efficiency are given. The minimum flow rate required is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water levels at the reservoir and the discharge site remain constant. **3** We assume the flow to be *frictionless* since the *minimum* flow rate is to be determined, $\dot{E}_{\text{mech,loss}} = 0$.



Properties We take the density of water to be ρ = 62.4 lbm/ft^3.

Analysis We take point 1 at the free surface of the reservoir and point 2 at the free surface of the discharge water stream, which is also taken as the reference level ($z_2 = 0$). Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$), the velocities are negligible at both points ($V_1 = V_2 = 0$), and frictional losses are disregarded. Then the energy equation in terms of heads for steady incompressible flow through a control volume between these two points that includes the turbine and the pipes reduces to

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

Substituting and noting that $\dot{W}_{\text{turbine, elect}} = \eta_{\text{turbine-gen}} \dot{m} g h_{\text{turbine, e}}$, the extracted turbine head and the mass and volume flow rates of water are determined to be

$$h_{\text{turbine, e}} = z_1 = 400 \text{ ft}$$

$$\dot{m} = \frac{\dot{W}_{\text{turbine,elect}}}{\eta_{\text{turbine-gen}} g h_{\text{turbine}}} = \frac{100 \text{ kW}}{0.85(32.2 \text{ ft/s}^2)(400 \text{ ft})} \left(\frac{25{,}037 \text{ ft}^2/\text{s}^2}{1 \text{ Btu/lbm}} \right) \left(\frac{0.9478 \text{ Btu/s}}{1 \text{ kW}} \right) = 216.8 \text{ lbm/s} \cong \mathbf{217 \text{ lbm/s}}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{216.8 \text{ lbm/s}}{62.4 \text{ lbm/ft}^3} = 3.47 \text{ ft}^3/\text{s}$$

Therefore, the flow rate of water must be at least 3.47 ft^3/s to generate the desired electric power while overcoming friction losses in pipes.

Discussion In an actual system, the flow rate of water will be more because of frictional losses in pipes.

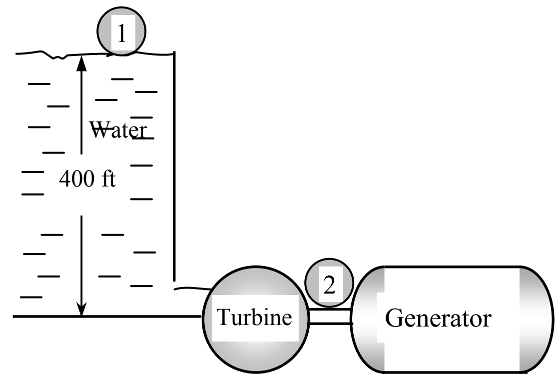
5-78E

Solution In a hydroelectric power plant, the elevation difference, the head loss, the power generation, and the overall turbine-generator efficiency are given. The flow rate required is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The water levels at the reservoir and the discharge site remain constant.

Properties We take the density of water to be ρ = 62.4 lbm/ft^3.

Analysis We take point 1 at the free surface of the reservoir and point 2 at the free surface of the discharge water stream, which is also taken as the reference level (z_2 = 0). Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$), the velocities are negligible at both points ($V_1 = V_2 = 0$). Then the energy equation in terms of heads for steady incompressible flow through a control volume between these two points that includes the turbine and the pipes reduces to



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

Substituting and noting that $\dot{W}_{\text{turbine, elect}}=\eta_{\text{turbine-gen}}\dot{m}gh_{\text{turbine, e}}$, the extracted turbine head and the mass and volume flow rates of water are determined to be

$$h_{\text{turbine, e}}=z_1-h_L=400-36=364\text{ ft}$$

$$\dot{m}=\frac{\dot{W}_{\text{turbine,elect}}}{\eta_{\text{turbine-gen}}gh_{\text{turbine}}}=\frac{100\text{ kW}}{0.85(32.2\text{ ft/s}^2)(364\text{ ft})}\left(\frac{25{,}037\text{ ft}^2/\text{s}^2}{1\text{ Btu/lbm}}\right)\left(\frac{0.9478\text{ Btu/s}}{1\text{ kW}}\right)=\mathbf{238\ lbm/s}$$

$$\dot{V}=\frac{\dot{m}}{\rho}=\frac{238\text{ lbm/s}}{62.4\text{ lbm/ft}^3}=3.82\text{ ft}^3/\text{s}$$

Therefore, the flow rate of water must be at least 3.82 ft^3/s to generate the desired electric power while overcoming friction losses in pipes.

Discussion Note that the effect of frictional losses in the pipes is to increase the required flow rate of water to generate a specified amount of electric power.

5-79

Solution A pump is pumping oil at a specified rate. The pressure rise of oil in the pump is measured, and the motor efficiency is specified. The mechanical efficiency of the pump is to be determined.

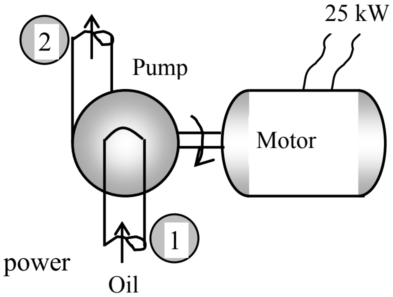
Assumptions **1** The flow is steady and incompressible. **2** The elevation difference across the pump is negligible. **3** All the losses in the pump are accounted for by the pump efficiency and thus $h_L = 0$. **4** The kinetic energy correction factors are given to be $\alpha_1 = \alpha_2 = \alpha = 1.05$.

Properties The density of oil is given to be $\rho = 860$ kg/m^3.

Analysis We take points 1 and 2 at the inlet and the exit of the pump, respectively. Noting that $z_1 = z_2$, the energy equation for the pump reduces to

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

where



$$V_1=\frac{\dot V}{A_1}=\frac{\dot V}{\pi D_1^2/4}=\frac{0.1\ \text{m}^3/\text{s}}{\pi(0.08\ \text{m})^2/4}=19.9\ \text{m/s}$$

$$V_2=\frac{\dot V}{A_2}=\frac{\dot V}{\pi D_2^2/4}=\frac{0.1\ \text{m}^3/\text{s}}{\pi(0.12\ \text{m})^2/4}=8.84\ \text{m/s}$$

Substituting, the useful pump head and the corresponding useful pumping power are determined to be

$$h_{\text{pump, u}}=\frac{250{,}000\ \text{N/m}^2}{(860\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)+\frac{1.05[(8.84\ \text{m/s})^2-(19.9\ \text{m/s})^2]}{2(9.81\ \text{m/s}^2)}=29.6-17.0=12.6\ \text{m}$$

$$\dot W_{\text{pump,u}}=\rho\dot V g h_{\text{pump, u}}=(860\ \text{kg/m}^3)(0.1\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(12.6\ \text{m})\left(\frac{1\ \text{kN}}{1000\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1\ \text{kN}\cdot\text{m/s}}\right)=10.6\ \text{kW}$$

Then the shaft pumping power and the mechanical efficiency of the pump become

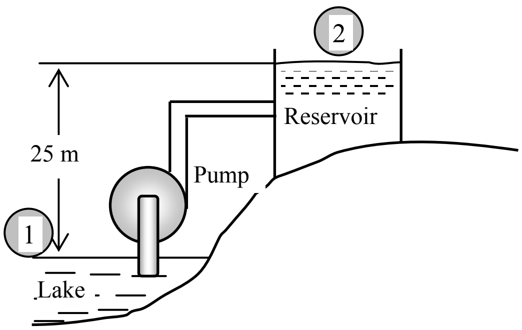
$$\dot W_{\text{pump,shaft}}=\eta_{\text{motor}}\dot W_{\text{electric}}=(0.90)(25\ \text{kW})=22.5\ \text{kW}$$

$$\eta_{\text{pump}}=\frac{\dot W_{\text{pump, u}}}{\dot W_{\text{pump, shaft}}}=\frac{10.6\ \text{kW}}{22.5\ \text{kW}}=0.471=\mathbf{47.1\%}$$

Discussion The overall efficiency of this pump/motor unit is the product of the mechanical and motor efficiencies, which is $0.9\times0.471=0.42$.

5-80

Solution Water is pumped from a large lake to a higher reservoir. The head loss of the piping system is given. The mechanical efficiency of the pump is to be determined.



Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the reservoir is constant.

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

Analysis We choose points 1 and 2 at the free surfaces of the lake and the reservoir, respectively, and take the surface of the lake as the reference level ($z_1 = 0$). Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho}+\alpha_1\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{\text{pump}}=\dot{m}\left(\frac{P_2}{\rho}+\alpha_2\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech,loss}} \rightarrow \dot{W}_{\text{pump,u}}=\dot{m}gz_2+\dot{E}_{\text{mech loss, piping}}$$

since, in the absence of a turbine, $\dot{E}_{\text{mech, loss}}=\dot{E}_{\text{mech loss, pump}}+\dot{E}_{\text{mech loss, piping}}$ and $\dot{W}_{\text{pump, u}}=\dot{W}_{\text{pump}}-\dot{E}_{\text{mech loss, pump}}$. Noting that $\dot{E}_{\text{mech loss, piping}}=\dot{m}gh_L$, the useful pump power is

$$\begin{aligned}\dot{W}_{\text{pump,u}}&=\dot{m}gz_2+\dot{m}gh_L=\rho\dot{V}g(z_2+h_L)\\&=(1000\text{ kg/m}^3)(0.025\text{ m}^3\text{/s})(9.81\text{ m/s}^2)[(25+5)\text{ m}]\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)\\&=7.36\text{ kN}\cdot\text{m/s}=7.36\text{ kW}\end{aligned}$$

Then the mechanical efficiency of the pump becomes

$$\eta_{\text{pump}}=\frac{\dot{W}_{\text{pump, u}}}{\dot{W}_{\text{shaft}}}=\frac{7.36\text{ kW}}{10\text{ kW}}=0.736=\mathbf{73.6\%}$$

Discussion A more practical measure of performance of the pump is the overall efficiency, which can be obtained by multiplying the pump efficiency by the motor efficiency.

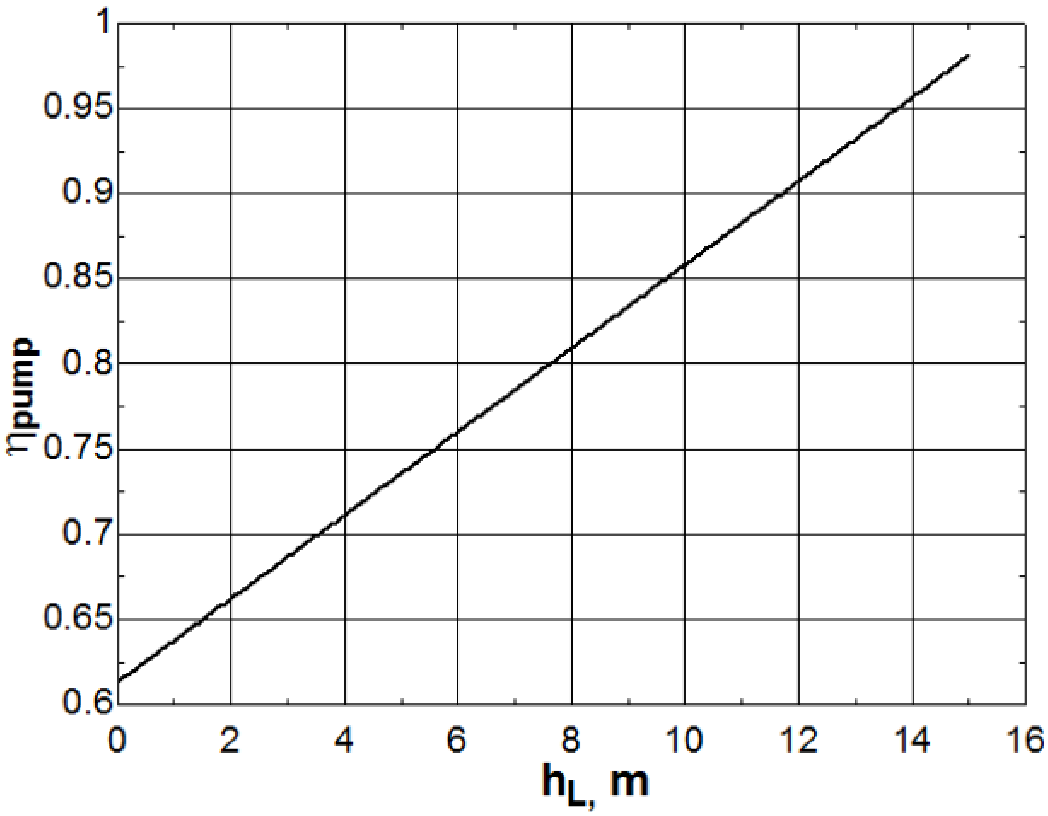
5-81

Solution The previous problem is reconsidered. The effect of head loss on mechanical efficiency of the pump. as the head loss varies 0 to 20 m in increments of 2 m is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
z2=25 "m"
W_shaft=10 "kW"
V_dot=0.025 "m3/s"
W_pump_u=rho*V_dot*g*(z2+h_L)/1000 "kW"
Eta_pump=W_pump_u/W_shaft
```

Head Loss, h_L, m	Pumping power $W_{pump, u}$	Efficiency η_{pump}
0	6.13	0.613
1	6.38	0.638
2	6.62	0.662
3	6.87	0.687
4	7.11	0.711
5	7.36	0.736
6	7.60	0.760
7	7.85	0.785
8	8.09	0.809
9	8.34	0.834
10	8.58	0.858
11	8.83	0.883
12	9.07	0.907
13	9.32	0.932
14	9.56	0.956
15	9.81	0.981



Discussion Note that the useful pumping power is used to raise the fluid and to overcome head losses. For a given power input, the pump that overcomes more head loss is more efficient.

5-82

Solution A pump with a specified shaft power and efficiency is used to raise water to a higher elevation. The maximum flow rate of water is to be determined.

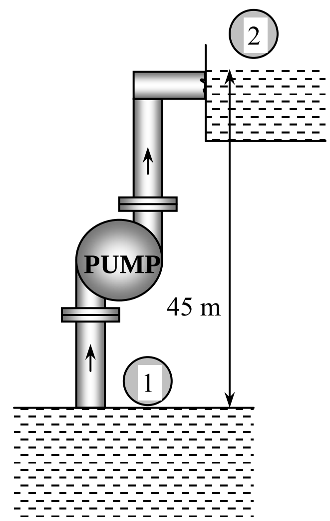
Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the reservoirs is constant. **3** We assume the flow in the pipes to be *frictionless* since the *maximum* flow rate is to be determined, $\dot{E}_{\text{mech loss, piping}} = 0$.

Properties We take the density of water to be $\rho = 1000\ \text{kg/m}^3$.

Analysis We choose points 1 and 2 at the free surfaces of the lower and upper reservoirs, respectively, and take the surface of the lower reservoir as the reference level ($z_1 = 0$). Both points are open to the atmosphere ($P_1 = P_2 = P_{\text{atm}}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho}+\alpha_1\frac{V_1^2}{2}+gz_1\right)+\dot{W}_{\text{pump}}=\dot{m}\left(\frac{P_2}{\rho}+\alpha_2\frac{V_2^2}{2}+gz_2\right)+\dot{W}_{\text{turbine}}+\dot{E}_{\text{mech,loss}} \quad\rightarrow\quad \dot{W}_{\text{pump,u}}=\dot{m}gz_2=\rho\dot{V}gz_2$$

since $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss, pump}}$ in this case and $\dot{W}_{\text{pump, u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss, pump}}$.



The useful pumping power is

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}}\dot{W}_{\text{pump, shaft}} = (0.82)(15\ \text{hp}) = 12.3\ \text{hp}$$

Substituting, the volume flow rate of water is determined to be

$$\dot{V}=\frac{\dot{W}_{\text{pump,u}}}{\rho g z_2}=\frac{12.3\ \text{hp}}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)(45\ \text{m})}\left(\frac{745.7\ \text{W}}{1\ \text{hp}}\right)\left(\frac{1\ \text{N}\cdot\text{m/s}}{1\ \text{W}}\right)\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)$$

$$= \mathbf{0.0208\ m^3/s}$$

Discussion This is the maximum flow rate since the frictional effects are ignored. In an actual system, the flow rate of water will be less because of friction in pipes.

5-83

Solution Water flows at a specified rate in a horizontal pipe whose diameter is decreased by a reducer. The pressures are measured before and after the reducer. The head loss in the reducer is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pipe is horizontal. **3** The kinetic energy correction factors are given to be $\alpha_1 = \alpha_2 = \alpha = 1.05$.

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

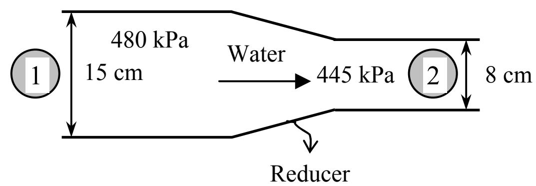
Analysis We take points 1 and 2 along the centerline of the pipe before and after the reducer, respectively. Noting that $z_1 = z_2$, the energy equation for steady incompressible flow through a control volume between these two points reduces to

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

where

$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.035 \text{ m}^3/\text{s}}{\pi(0.15 \text{ m})^2/4} = 1.981 \text{ m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.035 \text{ m}^3/\text{s}}{\pi(0.08 \text{ m})^2/4} = 6.963 \text{ m/s}$$



Substituting, the head loss in the reducer is determined to be

$$h_L = \frac{(480 - 445)\text{ kPa}}{(1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)}\left(\frac{1 \text{ kN/m}^2}{1 \text{ kPa}}\right)\left(\frac{1000 \text{ kg}\cdot\text{m/s}^2}{1 \text{ kN}}\right) + \frac{1.05[(1.981 \text{ m/s})^2 - (6.963 \text{ m/s})^2]}{2(9.81 \text{ m/s}^2)}$$
$$= 1.183 \text{ m} \cong \mathbf{1.18\ m}$$

Discussion Note that the 1.19 m of the head loss is due to frictional effects and 2.27 m is due to the increase in velocity. This head loss corresponds to a power potential loss of

$$\dot{E}_{\text{mech loss, piping}} = \rho \dot{V} g h_L = (1000 \text{ kg/m}^3)(0.035 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(1.19 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1 \text{ W}}{1 \text{ N}\cdot\text{m/s}}\right) = 406 \text{ W}$$

5-84

Solution A hose connected to the bottom of a tank is equipped with a nozzle at the end pointing straight up. The water is pressurized by a pump, and the height of the water jet is measured. The minimum pressure rise supplied by the pump is to be determined.

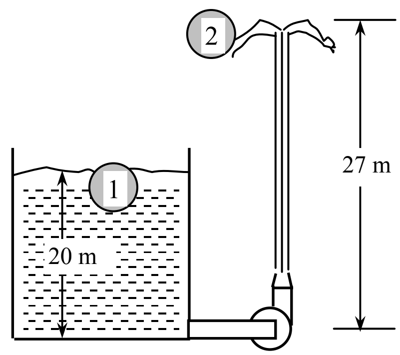
Assumptions **1** The flow is steady and incompressible. **2** Friction between the water and air as well as friction in the hose is negligible. **3** The water surface is open to the atmosphere.

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

Analysis We take point 1 at the free surface of water in the tank, and point 2 at the top of the water trajectory where $V_2 = 0$ and $P_1 = P_2 = P_{atm}$. Also, we take the reference level at the bottom of the tank. Noting that $z_1 = 20$ m and $z_2 = 27$ m, $h_L = 0$ (to get the minimum value for required pressure rise), and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the water stream reduces to

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

$$\rightarrow \quad h_{\text{pump, u}} = z_2 - z_1$$



Substituting,

$$h_{\text{pump, u}} = 27 - 20 = 7 \text{ m}$$

A water column height of 7 m corresponds to a pressure rise of

$$\Delta P_{\text{pump, min}} = \rho g h_{\text{pump, u}} = (1000 \text{ kg/m}^3)(9.81 \text{ m/s}^2)(7 \text{ m})\left(\frac{1 \text{ N}}{1000 \text{ kg} \cdot \text{m/s}^2}\right)$$

$$= 68.7 \text{ kN/m}^2 = \mathbf{68.7 \text{ kPa}}$$

Therefore, the pump must supply a minimum pressure rise of 68.7 kPa.

Discussion The result obtained above represents the minimum value, and should be interpreted accordingly. In reality, a larger pressure rise will need to be supplied to overcome friction.

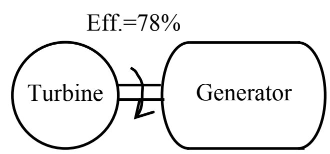
5-85

Solution The available head of a hydraulic turbine and its overall efficiency are given. The electric power output of this turbine is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The available head remains constant.

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

Analysis When the turbine head is available, the corresponding power output is determined from



$$\dot{W}_{\text{turbine}} = \eta_{\text{turbine}} \dot{m} g h_{\text{turbine}} = \eta_{\text{turbine}} \rho \dot{V} g h_{\text{turbine}}$$

Substituting,

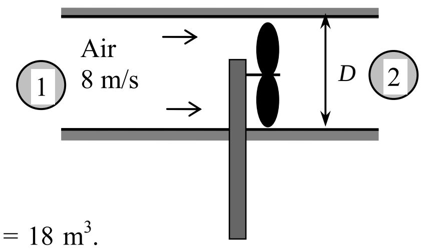
$$\dot{W}_{\text{turbine}} = 0.78(1000 \text{ kg/m}^3)(1.30 \text{ m}^3\text{/s})(9.81 \text{ m/s}^2)(50 \text{ m})\left(\frac{1 \text{ N}}{1 \text{ kg} \cdot \text{m/s}^2}\right)\left(\frac{1 \text{ kW}}{1000 \text{ N} \cdot \text{m/s}}\right) = \mathbf{497 \text{ kW}}$$

Discussion The power output of a hydraulic turbine is proportional to the available turbine head and the flow rate.

5-86

Solution A fan is to ventilate a bathroom by replacing the entire volume of air once every 10 minutes while air velocity remains below a specified value. The wattage of the fan-motor unit, the diameter of the fan casing, and the pressure difference across the fan are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Frictional losses along the flow (other than those due to the fan-motor inefficiency) are negligible. **3** The fan unit is horizontal so that z = constant along the flow (or, the elevation effects are negligible because of the low density of air). **4** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.



Properties The density of air is given to be 1.25 kg/m^3.

Analysis (*a*) The volume of air in the bathroom is V = 2 m × 3 m × 3 m = 18 m^3. Then the volume and mass flow rates of air through the casing must be

$$\dot{V} = \frac{V}{\Delta t} = \frac{18\text{ m}^3}{10 \times 60\text{ s}} = 0.03\text{ m}^3/\text{s}$$

$$\dot{m} = \rho \dot{V} = (1.25\text{ kg/m}^3)(0.03\text{ m}^3/\text{s}) = 0.0375\text{ kg/s}$$

We take points 1 and 2 on the inlet and exit sides of the fan, respectively. Point 1 is sufficiently far from the fan so that $P_1 = P_{atm}$ and the flow velocity is negligible ($V_1 = 0$). Also, $P_2 = P_{atm}$. Then the energy equation for this control volume between the points 1 and 2 reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1 \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{pump} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2 \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{turbine} + \dot{E}_{mech,loss} \quad \rightarrow \quad \dot{W}_{fan,u} = \dot{m}\alpha_2 \frac{V_2^2}{2}$$

since $\dot{E}_{mech,\,loss} = \dot{E}_{mech\,loss,\,pump}$ in this case and $\dot{W}_{pump,\,u} = \dot{W}_{pump} - \dot{E}_{mech\,loss,\,pump}$. Substituting,

$$\dot{W}_{fan,\,u} = \dot{m}\alpha_2 \frac{V_2^2}{2} = (0.0375\text{ kg/s})(1.0)\frac{(8\text{ m/s})^2}{2}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ W}}{1\text{ N}\cdot\text{m/s}}\right) = 1.2\text{ W}$$

and $$\dot{W}_{fan,\,elect} = \frac{\dot{W}_{fan,\,u}}{\eta_{fan\text{-}motor}} = \frac{1.2\text{ W}}{0.5} = \mathbf{2.4\ W}$$

Therefore, the electric power rating of the fan/motor unit must be 2.4 W.

(*b*) For air mean velocity to remain below the specified value, the diameter of the fan casing should be

$$\dot{V} = A_2 V_2 = (\pi D_2^2 / 4)V_2 \quad \rightarrow \quad D_2 = \sqrt{\frac{4\dot{V}}{\pi V_2}} = \sqrt{\frac{4(0.03\text{ m}^3/\text{s})}{\pi(8\text{ m/s})}} = 0.069\text{ m} = \mathbf{6.9\ cm}$$

(*c*) To determine the pressure difference across the fan unit, we take points 3 and 4 to be on the two sides of the fan on a horizontal line. Noting that $z_3 = z_4$ and $V_3 = V_4$ since the fan is a narrow cross-section and neglecting flow loses (other than the loses of the fan unit, which is accounted for by the efficiency), the energy equation for the fan section reduces to

$$\dot{m}\frac{P_3}{\rho} + \dot{W}_{fan,\,u} = \dot{m}\frac{P_4}{\rho} \quad \rightarrow \quad P_4 - P_3 = \frac{\dot{W}_{fan,\,u}}{\dot{m}/\rho} = \frac{\dot{W}_{fan,\,u}}{\dot{V}}$$

Substituting , $$P_4 - P_3 = \frac{1.2\text{ W}}{0.03\text{ m}^3/\text{s}}\left(\frac{1\text{ N}\cdot\text{m/s}}{1\text{ W}}\right) = 40\text{ N/m}^2 = \mathbf{40\ Pa}$$

Therefore, the fan will raise the pressure of air by 40 Pa before discharging it.

Discussion Note that only half of the electric energy consumed by the fan-motor unit is converted to the mechanical energy of air while the remaining half is converted to heat because of imperfections.

5-87

Solution Water flows through a horizontal pipe at a specified rate. The pressure drop across a valve in the pipe is measured. The corresponding head loss and the power needed to overcome it are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The pipe is given to be horizontal (otherwise the elevation difference across the valve is negligible). **3** The mean flow velocities at the inlet and the exit of the valve are equal since the pipe diameter is constant.

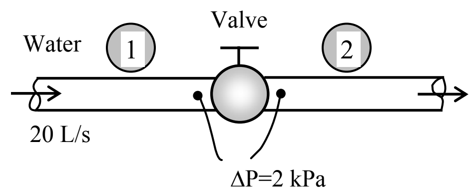
Properties We take the density of water to be $\rho = 1000$ kg/m^3.

Analysis We take the valve as the control volume, and points 1 and 2 at the inlet and exit of the valve, respectively. Noting that $z_1 = z_2$ and $V_1 = V_2$, the energy equation for steady incompressible flow through this control volume reduces to

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

Substituting,

$$h_L = \frac{2\ \text{kN/m}^2}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right) = \mathbf{0.204\ m}$$



The useful pumping power needed to overcome this head loss is

$$\dot{W}_{\text{pump, u}} = \dot{m} g h_L = \rho \dot{V} g h_L$$

$$= (1000\ \text{kg/m}^3)(0.020\ \text{m}^3/\text{s})(9.81\ \text{m/s}^2)(0.204\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{W}}{1\ \text{N}\cdot\text{m/s}}\right) = \mathbf{40\ W}$$

Therefore, this valve would cause a head loss of 0.204 m, and it would take 40 W of useful pumping power to overcome it.

Discussion The required useful pumping power could also be determined from

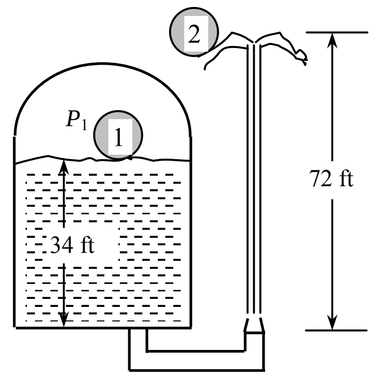
$$\dot{W}_{\text{pump}} = \dot{V}\Delta P = (0.020\ \text{m}^3/\text{s})(2000\ \text{Pa})\left(\frac{1\ \text{W}}{1\ \text{Pa}\cdot\text{m}^3/\text{s}}\right) = \mathbf{40\ W}$$

5-88E

Solution A hose connected to the bottom of a pressurized tank is equipped with a nozzle at the end pointing straight up. The minimum tank air pressure (gage) corresponding to a given height of water jet is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** Friction between water and air as well as friction in the hose is negligible. **3** The water surface is open to the atmosphere.

Properties We take the density of water to be $\rho = 62.4$ lbm/ft^3.



Analysis We take point 1 at the free surface of water in the tank, and point 2 at the top of the water trajectory where $V_2 = 0$ and $P_1 = P_2 = P_{atm}$. Also, we take the reference level at the bottom of the tank. Noting that $z_1 = 34$ ft and $z_2 = 72$ ft, $h_L = 0$ (to get the minimum value for the required air pressure), and that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$), the energy equation for steady incompressible flow through a control volume between these two points reduces to

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

or $$\frac{P_1 - P_{atm}}{\rho g} = z_2 - z_1 \rightarrow \frac{P_{1,gage}}{\rho g} = z_2 - z_1$$

Rearranging and substituting, the gage pressure of pressurized air in the tank is determined to be

$$P_{1,\text{gage}} = \rho g(z_2 - z_1) = (62.4 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)(72 - 34 \text{ ft})\left(\frac{1 \text{ lbf}}{32.2 \text{ lbm} \cdot \text{ft/s}^2}\right)\left(\frac{1 \text{ psi}}{144 \text{ lbf/ft}^2}\right) = \mathbf{16.5 \text{ psi}}$$

Therefore, the gage air pressure on top of the water tank must be at least 10.4 psi.

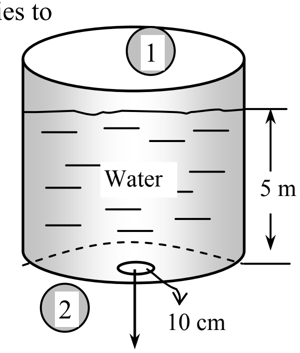
Discussion The result obtained above represents the minimum value, and should be interpreted accordingly. In reality, a larger pressure will be needed to overcome friction.

5-89

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere. The initial discharge velocity from the tank is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The tank is open to the atmosphere. **3** The kinetic energy correction factor at the orifice is given to be $\alpha_2 = \alpha = 1.2$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the orifice. 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 of the tank is very low ($V_1 \cong 0$), the energy equation between these two points (in terms of heads) simplifies to



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

which yields

$$z_1 + \alpha_2 \frac{V_2^2}{2g} = z_2 + h_L$$

Solving for V_2 and substituting,

$$V_2 = \sqrt{2g(z_1 - z_2 - h_L)/\alpha} = \sqrt{2(9.81 \text{ m/s}^2)(5 - 0.3 \text{ m})/1.2} = \mathbf{8.77 \text{ m/s}}$$

Discussion This is the velocity that will prevail at the beginning. The mean flow velocity will decrease as the water level in the tank decreases.

5-90

Solution Water enters a hydraulic turbine-generator system with a known flow rate, pressure drop, and efficiency. The net electric power output is to be determined.

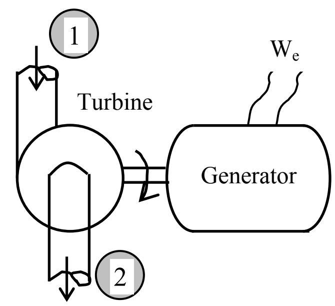
Assumptions **1** The flow is steady and incompressible. **2** All losses in the turbine are accounted for by turbine efficiency and thus $h_L = 0$. **3** The elevation difference across the turbine is negligible. **4** The effect of the kinetic energy correction factors is negligible, $\alpha_1 = \alpha_2 = \alpha = 1$.

Properties We take the density of water to be 1000 kg/m^3 and the density of mercury to be 13,560 kg/m^3.

Analysis We choose points 1 and 2 at the inlet and the exit of the turbine, respectively. Noting that the elevation effects are negligible, the energy equation in terms of heads for the turbine reduces to

$$\frac{P_1}{\rho g}+\alpha_1\frac{V_1^2}{2g}+z_1+h_{\text{pump, u}}=\frac{P_2}{\rho g}+\alpha_2\frac{V_2^2}{2g}+z_2+h_{\text{turbine, e}}+h_L \quad\rightarrow\quad h_{\text{turbine, e}}=\frac{P_1-P_2}{\rho_{\text{water}}g}+\frac{\alpha(V_1^2-V_2^2)}{2g} \qquad (1)$$

where



$$V_1=\frac{\dot V}{A_1}=\frac{\dot V}{\pi D_1^2/4}\frac{0.6\text{ m}^3/\text{s}}{\pi(0.30\text{ m})^2/4}=8.49\text{ m/s}$$

$$V_2=\frac{\dot V}{A_2}=\frac{\dot V}{\pi D_2^2/4}=\frac{0.6\text{ m}^3/\text{s}}{\pi(0.25\text{ m})^2/4}=12.2\text{ m/s}$$

The pressure drop corresponding to a differential height of 1.2 m in the mercury manometer is

$$P_1-P_2=(\rho_{\text{Hg}}-\rho_{\text{water}})gh$$
$$=[(13{,}560-1000)\text{ kg/m}^3](9.81\text{ m/s}^2)(1.2\text{ m})\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right)$$
$$=148\text{ kN/m}^2=148\text{ kPa}$$

Substituting into Eq. (1), the turbine head is determined to be

$$h_{\text{turbine, e}}=\frac{148\text{ kN/m}^2}{(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right)+(1.0)\frac{(8.49\text{ m/s})^2-(12.2\text{ m/s})^2}{2(9.81\text{ m/s}^2)}=15.1-3.9=11.2\text{ m}$$

Then the net electric power output of this hydroelectric turbine becomes

$$\dot W_{\text{turbine}}=\eta_{\text{turbine-gen}}\dot m g h_{\text{turbine, e}}=\eta_{\text{turbine-gen}}\rho\dot V g h_{\text{turbine, e}}$$
$$=0.83(1000\text{ kg/m}^3)(0.6\text{ m}^3/\text{s})(9.81\text{ m/s}^2)(11.2\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right)=\mathbf{55\text{ kW}}$$

Discussion It appears that this hydroelectric turbine will generate 55 kW of electric power under given conditions. Note that almost half of the available pressure head is discarded as kinetic energy. This demonstrates the need for a larger turbine exit area and better recovery. For example, the power output can be increased to 74 kW by redesigning the turbine and making the exit diameter of the pipe equal to the inlet diameter, $D_2 = D_1$. Further, if a much larger exit diameter is used and the exit velocity is reduced to a very low level, the power generation can increase to as much as 92 kW.

5-91

Solution The velocity profile for turbulent flow in a circular pipe is given. The kinetic energy correction factor for this flow is to be determined.

Analysis The velocity profile is given by $u(r) = u_{\max}(1 - r/R)^{1/n}$ with n = 9 The kinetic energy correction factor is then expressed as

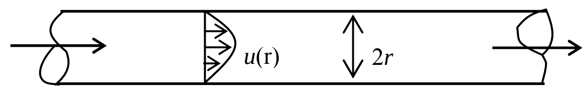
$$\alpha = \frac{1}{A}\int_A \left(\frac{u(r)}{V_{avg}}\right)^3 dA = \frac{1}{AV_{avg}^3}\int_A u(r)^3\, dA = \frac{1}{\pi R^2 V_{avg}^3}\int_{r=0}^{R} u_{\max}^3 \left(1-\frac{r}{R}\right)^{\frac{3}{n}}(2\pi r)dr = \frac{2u_{\max}^3}{R^2 V_{avg}^3}\int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{\frac{3}{n}} rdr$$

where the average velocity is

$$V_{avg} = \frac{1}{A}\int_A u(r)dA = \frac{1}{\pi R^2}\int_{r=0}^{R} u_{\max}\left(1-\frac{r}{R}\right)^{1/n}(2\pi r)dr = \frac{2u_{\max}}{R^2}\int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{1/n} rdr$$

From integral tables,

$$\int (a+bx)^n\, xdx = \frac{(a+bx)^{n+2}}{b^2(n+2)} - \frac{a(a+bx)^{n+1}}{b^2(n+1)}$$



Then,

$$\int_{r=0}^{R} u(r) rdr = \int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{1/n} rdr = \left.\frac{(1-r/R)^{\frac{1}{n}+2}}{\frac{1}{R^2}(\frac{1}{n}+2)} - \frac{(1-r/R)^{\frac{1}{n}+1}}{\frac{1}{R^2}(\frac{1}{n}+1)}\right|_{r=0}^{R} = \frac{n^2R^2}{(n+1)(2n+1)}$$

$$\int_{r=0}^{R} u(r)^3 rdr = \int_{r=0}^{R}\left(1-\frac{r}{R}\right)^{3/n} rdr = \left.\frac{(1-r/R)^{\frac{3}{n}+2}}{\frac{1}{R^2}(\frac{3}{n}+2)} - \frac{(1-r/R)^{\frac{3}{n}+1}}{\frac{1}{R^2}(\frac{3}{n}+1)}\right|_{r=0}^{R} = \frac{n^2R^2}{(n+3)(2n+3)}$$

Substituting,

$$V_{avg} = \frac{2u_{\max}}{R^2}\frac{n^2R^2}{(n+1)(2n+1)} = \frac{2n^2 u_{\max}}{(n+1)(2n+1)} = 0.8167u_{\max}$$

and

$$\alpha = \frac{2u_{\max}^3}{R^2}\left(\frac{2n^2 u_{\max}}{(n+1)(2n+1)}\right)^{-3}\frac{n^2R^2}{(n+3)(2n+3)} = \frac{(n+1)^3(2n+1)^3}{4n^4(n+3)(2n+3)} = \frac{(9+1)^3(2\times 9+1)^3}{4\times 9^4(9+3)(2\times 9+3)} = 1.037 \cong \mathbf{1.04}$$

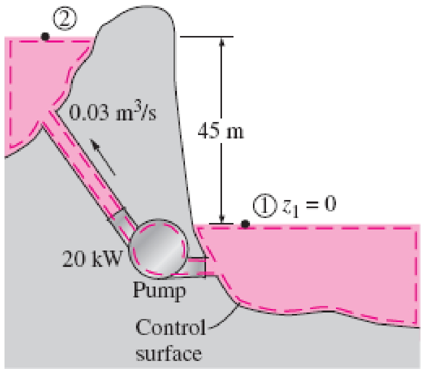
Discussion Note that ignoring the kinetic energy correction factor results in an error of just 4% in this case in the kinetic energy term (which may be small itself). Considering that the uncertainties in some terms are usually more that 4%, we can usually ignore this correction factor in turbulent pipe flow analyses. However, for laminar pipe flow analyses, α is equal to 2.0 for fully developed laminar pipe flow, and ignoring α may lead to significant errors.

5-92

Solution Water is pumped from a lower reservoir to a higher one. The head loss and power loss associated with this process are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the reservoirs is constant.

Properties We take the density of water to be 1000 kg/m³.



Analysis The mass flow rate of water through the system is

$$\dot{m} = \rho \dot{V} = (1000\ \text{kg/m}^3)(0.030\ \text{m}^3/\text{s}) = 30\ \text{kg/s}$$

We choose points 1 and 2 at the free surfaces of the lower and upper reservoirs, respectively, and take the surface of the lower reservoir as the reference level ($z_1 = 0$). Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow for a control volume between 1 and 2 reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1 \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump, u}} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2 \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine, e}} + \dot{E}_{\text{mech,loss}}$$

$$\dot{W}_{\text{pump, u}} = \dot{m}gz_2 + \dot{E}_{\text{mech,loss}} \quad \rightarrow \quad \dot{E}_{\text{mech,loss}} = \dot{W}_{\text{pump, u}} - \dot{m}gz_2$$

Substituting, the lost mechanical power and head loss are calculated as

$$\dot{E}_{\text{mech,loss}} = 20\ \text{kW} - (30\ \text{kg/s})(9.81\ \text{m/s}^2)(45\ \text{m})\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kW}}{1000\ \text{N}\cdot\text{m/s}}\right) = \mathbf{6.76\ kW}$$

Noting that the entire mechanical losses are due to frictional losses in piping and thus $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss, piping}}$, the irreversible head loss is determined to be

$$h_L = \frac{\dot{E}_{\text{mech loss, piping}}}{\dot{m}g} = \frac{6.76\ \text{kW}}{(30\ \text{kg/s})(9.81\ \text{m/s}^2)}\left(\frac{1\ \text{kg}\cdot\text{m/s}^2}{1\ \text{N}}\right)\left(\frac{1000\ \text{N}\cdot\text{m/s}}{1\ \text{kW}}\right) = \mathbf{23.0\ m}$$

Discussion The 6.76 kW of power is used to overcome the friction in the piping system. Note that the pump could raise the water an additional 23 m if there were no irreversible head losses in the system. In this ideal case, the pump would function as a turbine when the water is allowed to flow from the upper reservoir to the lower reservoir and extract 20 kW of power from the water.

5-93

Solution Water from a pressurized tank is supplied to a roof top. The discharge rate of water from the tank is to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The effect of the kinetic energy correction factor is negligible and thus $\alpha_2 = 1$ (we examine the effect of this approximation in the discussion).

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

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the discharge pipe. Noting that the fluid velocity at the free surface of the tank is very low ($V_1 \cong 0$) and water discharges into the atmosphere (and thus $P_2 = P_{atm}$), the energy equation written in the head form simplifies to

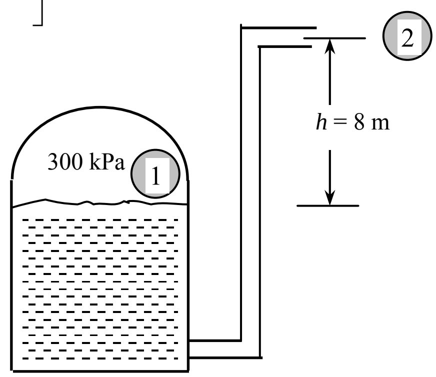
$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 + h_{\text{pump, u}} = \frac{P_2}{\rho g} + \alpha_2 \frac{V_2^2}{2g} + z_2 + h_{\text{turbine, e}} + h_L \quad \rightarrow \quad \frac{P_1 - P_{atm}}{\rho g} = \alpha_2 \frac{V_2^2}{2g} + z_2 - z_1 + h_L$$

Solving for V_2 and substituting, the discharge velocity is determined to

$$V_2 = \sqrt{\frac{1}{\alpha_2}\left[\frac{2P_{1,\text{ gage}}}{\rho} - 2g(z_2 - z_1 + h_L)\right]}$$

$$= \sqrt{\frac{1}{1}\left[\frac{2\times(300 \text{ kPa})}{1000 \text{ kg/m}^3}\left(\frac{1000 \text{ N/m}^2}{1 \text{ kPa}}\right)\left(\frac{1 \text{ kg}\cdot\text{m/s}^2}{1 \text{ N}}\right) - 2(9.81 \text{ m/s}^2)(8+2 \text{ m})\right]}$$

$$= 20.095 \text{ m/s} \cong 20.1 \text{ m/s}$$



Then the initial rate of discharge of water becomes

$$\dot{V} = A_{\text{orifice}} V_2 = \frac{\pi D^2}{4} V_2 = \frac{\pi (0.025 \text{ m})^2}{4}(20.095 \text{ m/s})$$

$$= 0.009864 \text{ m}^3/\text{s} \cong \mathbf{0.00986\ m^3/s = 9.86\ L/s}$$

Discussion This is the discharge rate that will prevail at the beginning. The mean flow velocity will decrease as the water level in the tank decreases. If we assume that the flow in the hose at the discharge is fully developed and turbulent, $\alpha_2 \approx 1.05$, and the results change to V_2 = 19.610 m/s ≈ 19.6 m/s, and $\dot{V} = 0.0096263 \text{ m}^3/\text{s} \cong \mathbf{0.00963\ m^3/s = 9.63\ L/s}$, a *decrease* (as expected since we are accounting for more losses) of about 2.4%.

5-94

Solution Underground water is pumped to a pool at a given elevation. The maximum flow rate and the pressures at the inlet and outlet of the pump are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the inlet and the outlet of the pump is negligible. **3** We assume the frictional effects in piping to be negligible since the *maximum* flow rate is to be determined, $\dot{E}_{\text{mech loss, pipping}} = 0$. **4** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

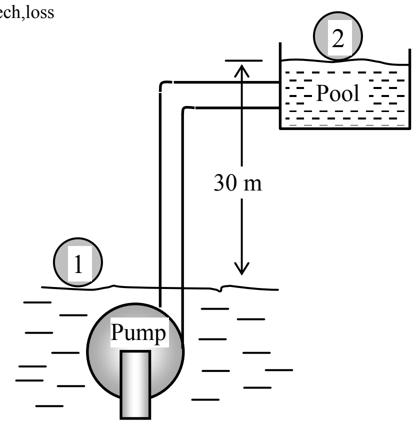
Properties We take the density of water to be 1 kg/L = 1000 kg/m^3.

Analysis (*a*) The pump-motor draws 5-kW of power, and is 78% efficient. Then the useful mechanical (shaft) power it delivers to the fluid is

$$\dot{W}_{\text{pump, u}} = \eta_{\text{pump-motor}} \dot{W}_{\text{electric}} = (0.78)(5\text{ kW}) = 3.9\text{ kW}$$

We take point 1 at the free surface of underground water, which is also taken as the reference level ($z_1 = 0$), and point 2 at the free surface of the pool. Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$), the velocities are negligible at both points ($V_1 \cong V_2 \cong 0$), and frictional losses in piping are disregarded. Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1 \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2 \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}}$$



In the absence of a turbine, $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss, pump}} + \dot{E}_{\text{mech loss, piping}}$ and

$\dot{W}_{\text{pump, u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss, pump}}$.

Thus, $\dot{W}_{\text{pump, u}} = \dot{m} g z_2$.

Then the mass and volume flow rates of water become

$$\dot{m} = \frac{\dot{W}_{\text{pump, u}}}{gz_2} = \frac{3.9\text{ kJ/s}}{(9.81\text{ m/s}^2)(30\text{ m})}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ}}\right) = 13.25\text{ kg/s}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{13.25\text{ kg/s}}{1000\text{ kg/m}^3} = 0.01325\text{ m}^3/\text{s} \cong \mathbf{0.0133\ m^3/s}$$

(*b*) We take points 3 and 4 at the inlet and the exit of the pump, respectively, where the flow velocities are

$$V_3 = \frac{\dot{V}}{A_3} = \frac{\dot{V}}{\pi D_3^2/4} = \frac{0.01325\text{ m}^3/\text{s}}{\pi(0.07\text{ m})^2/4} = 3.443\text{ m/s}, \quad V_4 = \frac{\dot{V}}{A_4} = \frac{\dot{V}}{\pi D_4^2/4} = \frac{0.01325\text{ m}^3/\text{s}}{\pi(0.05\text{ m})^2/4} = 6.748\text{ m/s}$$

We take the pump as the control volume. Noting that $z_3 = z_4$, the energy equation for this control volume reduces to

$$\dot{m}\left(\frac{P_3}{\rho} + \alpha_3 \frac{V_3^2}{2} + gz_3\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_4}{\rho} + \alpha_4 \frac{V_4^2}{2} + gz_4\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech loss, pump}} \rightarrow P_4 - P_3 = \frac{\rho\alpha(V_3^2 - V_4^2)}{2} + \frac{\dot{W}_{\text{pump,u}}}{\dot{V}}$$

Substituting,

$$P_4 - P_3 = \frac{(1000\text{ kg/m}^3)[(3.443\text{ m/s})^2 - (6.748\text{ m/s})^2]}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) + \frac{3.9\text{ kJ/s}}{0.01325\text{ m}^3/\text{s}}\left(\frac{1\text{ kN}\cdot\text{m}}{1\text{ kJ}}\right)$$

$$= (-16.8 + 294.3)\text{ kN/m}^2 = 277.5\text{ kPa} \cong \mathbf{278\ kPa}$$

Discussion In an actual system, the flow rate of water will be less because of friction in the pipes. Also, the effect of flow velocities on the pressure change across the pump is negligible in this case (under 2%) and can be ignored.

5-95

Solution Underground water is pumped to a pool at a given elevation. For a given head loss, the flow rate and the pressures at the inlet and outlet of the pump are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the inlet and the outlet of the pump is negligible. **3** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Properties We take the density of water to be 1 kg/L = 1000 kg/m^3.

Analysis (*a*) The pump-motor draws 5-kW of power, and is 78% efficient. Then the useful mechanical (shaft) power it delivers to the fluid is

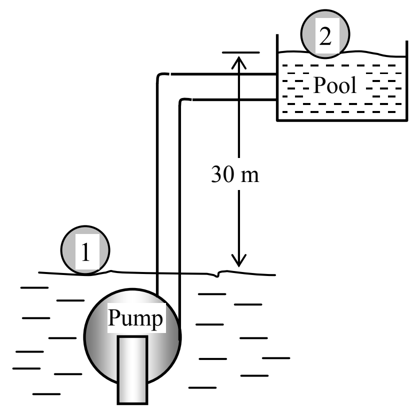
$$\dot{W}_{\text{pump, u}} = \eta_{\text{pump-motor}} \dot{W}_{\text{electric}} = (0.78)(5\text{ kW}) = 3.9\text{ kW}$$

We take point 1 at the free surface of underground water, which is also taken as the reference level ($z_1 = 0$), and point 2 at the free surface of the pool. Also, both 1 and 2 are open to the atmosphere ($P_1 = P_2 = P_{atm}$), and the velocities are negligible at both points ($V_1 \cong V_2 \cong 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

$$\dot{m}\left(\frac{P_1}{\rho} + \alpha_1 \frac{V_1^2}{2} + gz_1\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_2}{\rho} + \alpha_2 \frac{V_2^2}{2} + gz_2\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech,loss}}$$

In the absence of a turbine, $\dot{E}_{\text{mech, loss}} = \dot{E}_{\text{mech loss, pump}} + \dot{E}_{\text{mech loss, piping}}$ and $\dot{W}_{\text{pump, u}} = \dot{W}_{\text{pump}} - \dot{E}_{\text{mech loss, pump}}$ and thus

$$\dot{W}_{\text{pump, u}} = \dot{m} g z_2 + \dot{E}_{\text{mech loss, piping}}$$



Noting that $\dot{E}_{\text{mech, loss}} = \dot{m} g h_L$, the mass and volume flow rates of water become

$$\dot{m} = \frac{\dot{W}_{\text{pump, u}}}{gz_2 + gh_L} = \frac{\dot{W}_{\text{pump, u}}}{g(z_2 + h_L)}$$

$$= \frac{3.9\text{ kJ/s}}{(9.81\text{ m/s}^2)(30+4\text{ m})}\left(\frac{1000\text{ m}^2/\text{s}^2}{1\text{ kJ}}\right) = 11.69\text{ kg/s}$$

$$\dot{V} = \frac{\dot{m}}{\rho} = \frac{11.69\text{ kg/s}}{1000\text{ kg/m}^3} = 0.01169\text{ m}^3/\text{s} \cong \mathbf{0.0117\ m^3/s}$$

(*b*) We take points 3 and 4 at the inlet and the exit of the pump, respectively, where the flow velocities are

$$V_3 = \frac{\dot{V}}{A_3} = \frac{\dot{V}}{\pi D_3^2/4} = \frac{0.01169\text{ m}^3/\text{s}}{\pi(0.07\text{ m})^2/4} = 3.038\text{ m/s}, \quad V_4 = \frac{\dot{V}}{A_4} = \frac{\dot{V}}{\pi D_4^2/4} = \frac{0.01169\text{ m}^3/\text{s}}{\pi(0.05\text{ m})^2/4} = 5.954\text{ m/s}$$

We take the pump as the control volume. Noting that $z_3 = z_4$, the energy equation for this control volume reduces to

$$\dot{m}\left(\frac{P_3}{\rho} + \alpha_3 \frac{V_3^2}{2} + gz_3\right) + \dot{W}_{\text{pump}} = \dot{m}\left(\frac{P_4}{\rho} + \alpha_4 \frac{V_4^2}{2} + gz_4\right) + \dot{W}_{\text{turbine}} + \dot{E}_{\text{mech loss, pump}} \rightarrow P_4 - P_3 = \frac{\rho\alpha(V_3^2 - V_4^2)}{2} + \frac{\dot{W}_{\text{pump,u}}}{\dot{V}}$$

Substituting,

$$P_4 - P_3 = \frac{(1000\text{ kg/m}^3)[(3.038\text{ m/s})^2 - (5.954\text{ m/s})^2]}{2}\left(\frac{1\text{ kN}}{1000\text{ kg}\cdot\text{m/s}^2}\right) + \frac{3.9\text{ kJ/s}}{0.01169\text{ m}^3/\text{s}}\left(\frac{1\text{ kN}\cdot\text{m}}{1\text{ kJ}}\right)$$

$$= (-13.1 + 333.6)\text{ kN/m}^2 = 320.5\text{ kPa} \cong \mathbf{321\ kPa}$$

Discussion Note that frictional losses in the pipes causes the flow rate of water to decrease. Also, the effect of flow velocities on the pressure change across the pump is negligible in this case (about 1%) and can be ignored.

5-96E

Solution Water is pumped from a lake to a nearby pool by a pump with specified power and efficiency. The head loss of the piping system and the mechanical power used to overcome it are to be determined.

Assumptions **1** The flow is steady and incompressible. **2** The elevation difference between the lake and the free surface of the pool is constant. **3** All the losses in the pump are accounted for by the pump efficiency and thus h_L represents the losses in piping.

Properties We take the density of water to be $\rho = 62.4$ lbm/ft^3.

Analysis The useful pumping power and the corresponding useful pumping head are

$$\dot{W}_{\text{pump,u}} = \eta_{\text{pump}}\dot{W}_{\text{pump}} = (0.73)(12\text{ hp}) = 8.76\text{ hp}$$

$$h_{\text{pump, u}} = \frac{\dot{W}_{\text{pump, u}}}{\dot{m}g} = \frac{\dot{W}_{\text{pump, u}}}{\rho\dot{V}g}$$

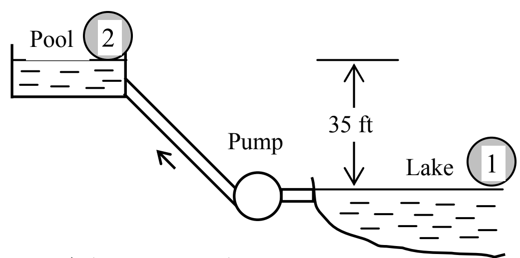
$$= \frac{8.76\text{ hp}}{(62.4\text{ lbm/ft}^3)(1.2\text{ ft}^3\text{/s})(32.2\text{ ft/s}^2)}\left(\frac{32.2\text{ lbm}\cdot\text{ft/s}^2}{1\text{ lbf}}\right)\left(\frac{550\text{ lbf}\cdot\text{ft/s}}{1\text{ hp}}\right) = 64.3\text{ ft}$$

We choose points 1 and 2 at the free surfaces of the lake and the pool, respectively. Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation for steady incompressible flow through a control volume between these two points that includes the pump and the pipes reduces to

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

Substituting, the head loss is determined to be

$$h_L = h_{\text{pump, u}} - (z_2 - z_1) = 64.3 - 35 = \mathbf{29.3\text{ ft}}$$



Then the power used to overcome it becomes

$$\dot{E}_{\text{mech loss, piping}} = \rho\dot{V}gh_L$$

$$= (62.4\text{ lbm/ft}^3)(1.2\text{ ft}^3\text{/s})(32.2\text{ ft/s}^2)(29.3\text{ ft})\left(\frac{1\text{ lbf}}{32.2\text{ lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\text{ hp}}{550\text{ lbf}\cdot\text{ft/s}}\right)$$

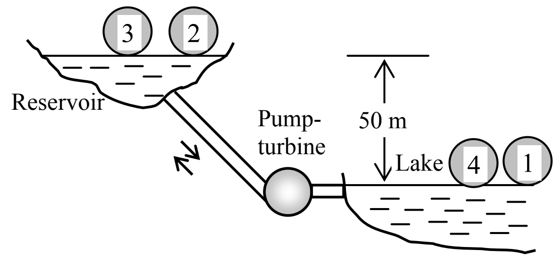
$$= \mathbf{4.0\text{ hp}}$$

Discussion Note that the pump must raise the water an additional height of 29.3 ft to overcome the frictional losses in pipes, which requires an additional useful pumping power of about 4 hp.

5-97

Solution An entrepreneur is to build a large reservoir above the lake level, and pump water from the lake to the reservoir at night using cheap power, and let the water flow from the reservoir back to the lake during the day, producing power. The potential revenue this system can generate per year is to be determined.

Assumptions **1** The flow in each direction is steady and incompressible. **2** The elevation difference between the lake and the reservoir can be taken to be constant, and the elevation change of reservoir during charging and discharging is disregarded. **3** The given unit prices remain constant. **4** The system operates every day of the year for 10 hours in each mode.



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

Analysis We choose points 1 and 2 at the free surfaces of the lake and the reservoir, respectively, and take the surface of the lake as the reference level. Both points are open to the atmosphere ($P_1 = P_2 = P_{atm}$) and the velocities at both locations are negligible ($V_1 = V_2 = 0$). Then the energy equation in terms of heads for steady incompressible flow through a control volume between these two points that includes the pump (or the turbine) and the pipes reduces to

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

$$h_{\text{pump, u}}=z_2+h_L=50+4=54\text{ m}$$

Turbine mode: (switch points 1 and 2 so that 1 is on inlet side) $\rightarrow$ $h_{\text{turbine, e}}=z_1-h_L=50-4=46\text{ m}$

The pump and turbine power corresponding to these heads are

$$\dot{W}_{\text{pump, elect}}=\frac{\dot{W}_{\text{pump, u}}}{\eta_{\text{pump-motor}}}=\frac{\rho\dot{V}gh_{\text{pump, u}}}{\eta_{\text{pump-motor}}}$$
$$=\frac{(1000\text{ kg/m}^3)(2\text{ m}^3\text{/s})(9.81\text{ m/s}^2)(54\text{ m})}{0.75}\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right)=1413\text{ kW}$$

$$\dot{W}_{\text{turbine}}=\eta_{\text{turbine-gen}}\dot{m}gh_{\text{turbine, e}}=\eta_{\text{turbine-gen}}\rho\dot{V}gh_{\text{turbine, e}}$$
$$=0.75(1000\text{ kg/m}^3)(2\text{ m}^3\text{/s})(9.81\text{ m/s}^2)(46\text{ m})\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kW}}{1000\text{ N}\cdot\text{m/s}}\right)=677\text{ kW}$$

Then the power cost of the pump, the revenue generated by the turbine, and the net income (revenue minus cost) per year become

$$\text{Cost}=\dot{W}_{\text{pump, elect}}\Delta t\times\text{Unit price}=(1413\text{ kW})(365\times 10\text{ h/year})(\$0.06\text{/kWh})=\$309{,}447\text{/year}$$

$$\text{Revenue}=\dot{W}_{\text{turbine}}\Delta t\times\text{Unit price}=(677\text{ kW})(365\times 10\text{ h/year})(\$0.13\text{/kWh})=\$321{,}237\text{/year}$$

Net income = Revenue – Cost = 321,237 – 309,447 = \$11,790/year ≅ **\$11,800/year**

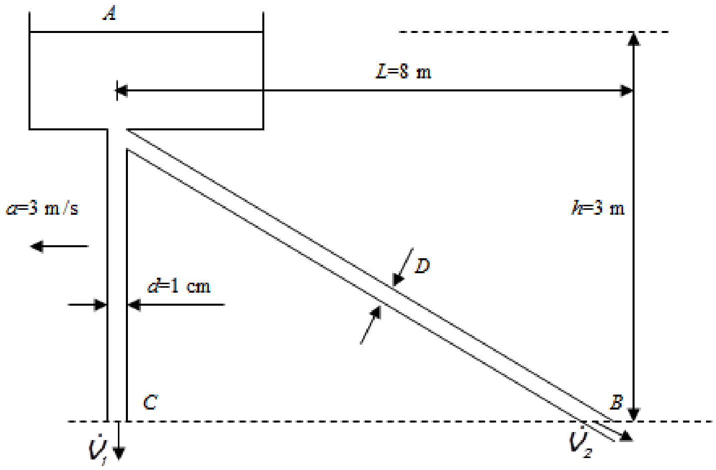
Discussion It appears that this pump-turbine system has a potential annual income of about \$11,800. A decision on such a system will depend on the initial cost of the system, its life, the operating and maintenance costs, the interest rate, and the length of the contract period, among other things.

5-98

Solution A tank with two discharge pipes accelerates to the left. The diameter of the inclined pipe is to be determined.

Assumptions 1 The flow is steady and incompressible. **2** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Analysis



Applying Bernoulli Equation between the points A and B

$$\left(\frac{P_A}{\gamma}+Z_A+\frac{W_A^2}{2g}\right)-\left(\frac{P_B}{\gamma}+Z_B+\frac{W_B^2}{2g}\right)=\frac{a}{g}\Delta L+Negligible\ \text{losses}$$

$$h=\frac{W_B^2}{2g}+\frac{a}{g}\Delta L \rightarrow \frac{W_B^2}{2g}=h-\frac{a}{g}\Delta L \qquad (1)$$

Applying Bernoulli Equation between the points A and C

$$h=\frac{W_C^2}{2g} \qquad (2)$$

Let's divide (1) by (2)

$$\frac{\frac{W_B^2}{2g}}{\frac{W_C^2}{2g}}=\frac{h-\frac{a}{g}\Delta L}{h} \text{ or } \left(\frac{W_B}{W_C}\right)^2=1-\frac{a\Delta L}{gh}, \quad \frac{W_B}{W_C}=\sqrt{1-\frac{a\Delta L}{gh}}$$

$$Q_B=Q_C \rightarrow A_B W_B=A_C W_C \rightarrow \frac{A_C}{A_B}=\frac{W_B}{W_C}=\frac{d^2}{D^2}$$

Therefore,

$$\frac{d^2}{D^2}=\sqrt{1-\frac{a\Delta L}{gh}}=\sqrt{1-\frac{3.18}{9,81.3}}=0.429$$

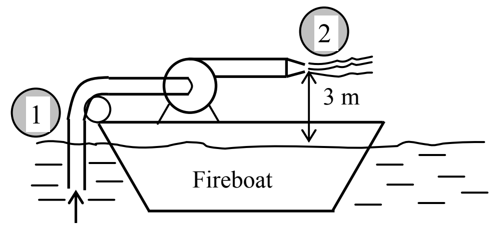
$$D^2=\frac{d^2}{0.429} \rightarrow D=\sqrt{\frac{0.01^2}{0.429^2}}\cong 1.53x10^{-2}\ m \rightarrow D\cong \mathbf{1.53\ cm}$$

5-99

Solution A fireboat is fighting fires by drawing sea water and discharging it through a nozzle. The head loss of the system and the elevation of the nozzle are given. The shaft power input to the pump and the water discharge velocity are to be determined.

Assumptions 1 The flow is steady and incompressible. **2** The effect of the kinetic energy correction factors is negligible, $\alpha = 1$.

Properties The density of sea water is given to be ρ =1030 kg/m^3.



Analysis We take point 1 at the free surface of the sea and point 2 at the nozzle exit. Noting that $P_1 = P_2 = P_{atm}$ and $V_1 \cong 0$ (point 1 is at the free surface; not at the pipe inlet), the energy equation for the control volume between 1 and 2 that includes the pump and the piping system reduces to

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

where the water discharge velocity is

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2 / 4} = \frac{0.04 \text{ m}^3/\text{s}}{\pi (0.05 \text{ m})^2 / 4} = 20.37 \text{ m/s} \cong \mathbf{20.4 \ m/s}$$

Substituting, the useful pump head and the corresponding useful pump power are determined to be

$$h_{pump,u} = (3 \text{ m}) + (1)\frac{(20.37 \text{ m/s})^2}{2(9.81 \text{ m/s}^2)} + (3 \text{ m}) = 27.15 \text{ m}$$

$$\dot{W}_{\text{pump,u}} = \rho \dot{V} g h_{pump,u}$$

$$= (1030 \text{ kg/m}^3)(0.1 \text{ m}^3/\text{s})(9.81 \text{ m/s}^2)(27.15 \text{ m})\left(\frac{1 \text{ kN}}{1000 \text{ kg} \cdot \text{m/s}^2}\right)\left(\frac{1 \text{ kW}}{1 \text{ kN} \cdot \text{m/s}}\right)$$

$$= 27.43 \text{ kW}$$

Then the required shaft power input to the pump becomes

$$\dot{W}_{\text{pump, shaft}} = \frac{\dot{W}_{\text{pump,u}}}{\eta_{\text{pump}}} = \frac{27.43 \text{ kW}}{0.70} = \mathbf{39.2 \ kW}$$

Discussion Note that the pump power is used primarily to increase the kinetic energy of water.

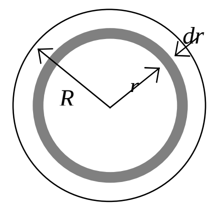
Review Problems

5-100

Solution A fluid is flowing in a circular pipe. A relation is to be obtained for the average fluid velocity in terms of $V(r)$, R, and r.

Analysis Choosing a circular ring of area $dA = 2\pi r dr$ as our differential area, the mass flow rate through a cross-sectional area can be expressed as

$$\dot{m} = \int_A \rho V(r) dA = \int_0^R \rho V(r) 2\pi r\, dr$$



Setting this equal to and solving for V_{avg},

$$\boxed{V_{avg} = \frac{2}{R^2}\int_0^R V(r)\, r\, dr}$$

Discussion If V were a function of both r and θ, we would also need to integrate with respect to θ.

5-101

Solution Air is accelerated in a nozzle. The density of air at the nozzle exit is to be determined.

Assumptions Flow through the nozzle is steady.

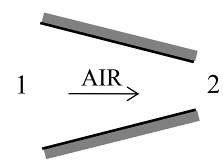
Properties The density of air is given to be 2.50 kg/m^3 at the inlet.

Analysis There is only one inlet and one exit, and thus $\dot{m}_1 = \dot{m}_2 = \dot{m}$. Then,

$$\dot{m}_1 = \dot{m}_2$$

$$\rho_1 A_1 V_1 = \rho_2 A_2 V_2$$

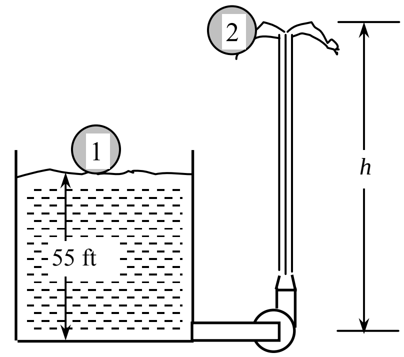
$$\rho_2 = \frac{A_1}{A_2}\frac{V_1}{V_2}\rho_1 = 2\frac{120\text{ m/s}}{330\text{ m/s}}(2.50\text{ kg/m}^3) = \mathbf{1.82\ kg/m^3}$$



Discussion Note that the density of air decreases considerably despite a decrease in the cross-sectional area of the nozzle.

5-102E

Solution A hose is connected to the bottom of a water tank open to the atmosphere. The hose is equipped with a pump and a nozzle at the end. The maximum height to which the water stream could rise is to be determined.



Assumptions **1** The flow is incompressible with negligible friction. **2** The friction between the water and air is negligible. **3** We take the head loss to be zero ($h_L = 0$) to determine the maximum rise of water jet.

Properties We take the density of water to be 62.4 lbm/ft^3.

Analysis We take point 1 at the free surface of the tank, and point 2 at the top of the water trajectory where $V_2 = 0$. We take the reference level at the bottom of the tank. 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 very low ($V_1 \cong 0$), the energy equation for a control volume between these two points (in terms of heads) simplifies to

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

where the useful pump head is

$$h_{\text{pump, u}} = \frac{\Delta P_{\text{pump}}}{\rho g} = \frac{10 \text{ psi}}{(62.4 \text{ lbm/ft}^3)(32.2 \text{ ft/s}^2)}\left(\frac{144 \text{ lbf/ft}^2}{1 \text{ psi}}\right)\left(\frac{32.2 \text{ lbm}\cdot\text{ft/s}^2}{1 \text{ lbf}}\right) = 23.1 \text{ ft}$$

Substituting, the maximum height rise of water jet from the ground level is determined to be

$$z_2 = z_1 + h_{\text{pump, u}} = 55 + 23.2 = \mathbf{78.2\ ft}$$

Discussion The actual rise of water will be less because of the frictional effects between the water and the hose walls and between the water jet and air.

5-103

Solution Water discharges from the orifice at the bottom of a pressurized tank. The time it will take for half of the water in the tank to be discharged and the water level after 10 s are to be determined.

Assumptions **1** The flow is incompressible, and the frictional effects are negligible. **2** The tank air pressure above the water level is maintained constant.

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

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of orifice. We take the positive direction of z to be upwards with reference level at the orifice ($z_2 = 0$). Fluid at point 2 is open to the atmosphere (and thus $P_2 = P_{atm}$) and the velocity at the free surface is very low ($V_1 \cong 0$). Then,

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}+z_1=\frac{P_{atm}}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{2gz_1+2P_{1,\text{gage}}/\rho}$$

or, $V_2=\sqrt{2gz+2P_{1,\text{gage}}/\rho}$ where z is the water height in the tank at any time t. Water surface moves down as the tank drains, and the value of z changes from H initially to 0 when the tank is emptied completely.

We denote the diameter of the orifice by D, and the diameter of the tank by D_o. The flow rate of water from the tank is obtained by multiplying the discharge velocity by the orifice cross-sectional area,

$$\dot{V}=A_{\text{orifice}}V_2=\frac{\pi D^2}{4}\sqrt{2gz+2P_{1,\text{gage}}/\rho}$$

Then the amount of water that flows through the orifice during a differential time interval dt is

$$dV=\dot{V}dt=\frac{\pi D^2}{4}\sqrt{2gz+2P_{1,\text{gage}}/\rho}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

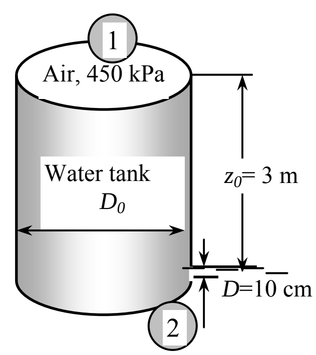
$$dV=A_{\text{tank}}(-dz)=-\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{2gz+2P_{1,\text{gage}}/\rho}\,dt=-\frac{\pi D_0^2}{4}dz \quad\rightarrow\quad dt=-\frac{D_0^2}{D^2}\sqrt{\frac{1}{2gz+2P_{1,\text{gage}}/\rho}}\,dz$$

The last relation can be integrated since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_0$ to $t = t$ when $z = z$ gives

$$\sqrt{\frac{2z_0}{g}+\frac{2P_{1,gage}}{\rho g^2}}-\sqrt{\frac{2z}{g}+\frac{2P_{1,gage}}{\rho g^2}}=\frac{D_0^2}{D^2}t$$



where $$\frac{2P_{1,\text{gage}}}{\rho g^2}=\frac{2(450-100)\ \text{kN/m}^2}{(1000\ \text{kg/m}^3)(9.81\ \text{m/s}^2)^2}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)=7.274\ \text{s}^2$$

The time for half of the water in the tank to be discharged ($z = z_0/2$) is

$$\sqrt{\frac{2(3\ \text{m})}{9.81\ \text{m/s}^2}+7.274\ \text{s}^2}-\sqrt{\frac{2(1.5\ \text{m})}{9.81\ \text{m/s}^2}+7.274\ \text{s}^2}=\frac{(0.1\ \text{m})^2}{(2\ \text{m})^2}t \quad\rightarrow\quad t = \mathbf{22.0\ s}$$

(b) Water level after 10 s is $$\sqrt{\frac{2(3\ \text{m})}{9.81\ \text{m/s}^2}+7.274\ \text{s}^2}-\sqrt{\frac{2z}{9.81\ \text{m/s}^2}+7.274\ \text{s}^2}=\frac{(0.1\ \text{m})^2}{(2\ \text{m})^2}(10\ \text{s})\rightarrow z = \mathbf{2.31\ m}$$

Discussion Note that the discharging time is inversely proportional to the square of the orifice diameter. Therefore, the discharging time can be reduced to one-fourth by doubling the diameter of the orifice.

5-104

Solution Air flows through a pipe that consists of two sections at a specified rate. The differential height of a water manometer placed between the two pipe sections is to be determined.

Assumptions **1**The flow through the pipe is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** The losses in the reducing section are negligible. **3** The pressure difference across an air column is negligible because of the low density of air, and thus the air column in the manometer can be ignored.

Properties The density of air is given to be $\rho_{air} = 1.20$ kg/m^3. We take the density of water to be $\rho_w = 1000$ kg/m^3.

Analysis We take points 1 and 2 along the centerline of the pipe over the two tubes of the manometer. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases), the Bernoulli equation between points 1 and 2 gives

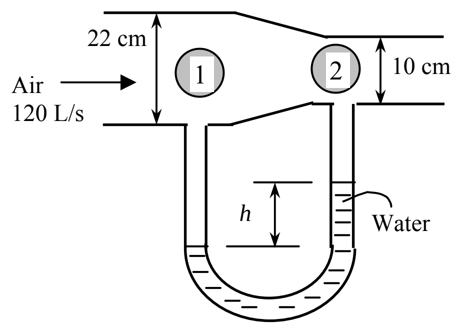
$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad P_1 - P_2 = \frac{\rho_{air}(V_2^2 - V_1^2)}{2} \qquad (1)$$

We let the differential height of the water manometer be h. Then the pressure difference $P_2 - P_1$ can also be expressed as

$$P_1 - P_2 = \rho_w g h \qquad (2)$$

Combining Eqs. (1) and (2) and solving for h,

$$\frac{\rho_{air}(V_2^2 - V_1^2)}{2} = \rho_w g h \quad \rightarrow \quad h = \frac{\rho_{air}(V_2^2 - V_1^2)}{2g\rho_w} = \frac{V_2^2 - V_1^2}{2g\rho_w / \rho_{air}}$$



Calculating the velocities and substituting,

$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.120\ \text{m}^3/\text{s}}{\pi(0.22\ \text{m})^2/4} = 3.157\ \text{m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.120\ \text{m}^3/\text{s}}{\pi(0.1\ \text{m})^2/4} = 15.28\ \text{m/s}$$

$$h = \frac{(15.28\ \text{m/s})^2 - (3.157\ \text{m/s})^2}{2(9.81\ \text{m/s})^2(1000/1.20)} = 0.01367\ \text{m} \cong \mathbf{1.37\ cm}$$

Therefore, the differential height of the water column will be 1.37 cm.

Discussion Note that the differential height of the manometer is inversely proportional to the density of the manometer fluid. Therefore, heavy fluids such as mercury are used when measuring large pressure differences.

5-105

Solution Air flows through a horizontal duct of variable cross-section. For a given differential height of a water manometer placed between the two pipe sections, the downstream velocity of air is to be determined, and an error analysis is to be conducted.

Assumptions **1** The flow through the duct is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** The losses in this section of the duct are negligible. **3** The pressure difference across an air column is negligible because of the low density of air, and thus the air column in the manometer can be ignored.

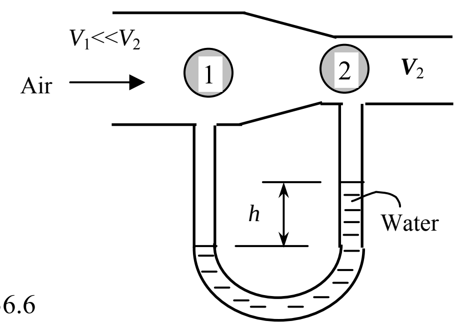
Properties The gas constant of air is R = 0.287 kPa·m^3/kg·K. We take the density of water to be ρ_w = 1000 kg/m^3.

Analysis We take points 1 and 2 along the centerline of the duct over the two tubes of the manometer. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases) and $V_1 \cong 0$, the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad\rightarrow\quad \frac{P_1}{\rho g}=\frac{P_2}{\rho g}+\frac{V_2^2}{2g} \quad\rightarrow\quad V_2=\sqrt{\frac{2(P_1-P_2)}{\rho_{air}}} \qquad (1)$$

where $P_1 - P_2 = \rho_w gh$

and $\rho_{air}=\frac{P}{RT}=\frac{100\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(298\text{ K})}=1.17\text{ kg/m}^3$



Substituting into (1), the downstream velocity of air V_2 is determined to be

$$V_2=\sqrt{\frac{2\rho_w gh}{\rho_{air}}}=\sqrt{\frac{2(1000\text{ kg/m}^3)(9.81\text{ m/s}^2)(0.08\text{ m})}{1.17\text{ kg/m}^3}}=\mathbf{36.6\ m/s} \qquad (2)$$

Therefore, the velocity of air increases from a low level in the first section to 36.6 m/s in the second section.

Error Analysis We observe from Eq. (2) that the velocity is proportional to the square root of the differential height of the manometer fluid. That is, $V_2 = k\sqrt{h}$.

Taking the differential: $dV_2 = \frac{1}{2}k\frac{dh}{\sqrt{h}}$

Dividing by V_2: $\frac{dV_2}{V_2}=\frac{1}{2}k\frac{dh}{\sqrt{h}}\frac{1}{k\sqrt{h}} \quad\rightarrow\quad \frac{dV_2}{V_2}=\frac{dh}{2h}=\frac{\pm 2\text{ mm}}{2\times 80\text{ mm}}=\mathbf{\pm\ 0.013}$

Therefore, the uncertainty in the velocity corresponding to an uncertainty of 2 mm in the differential height of water is 1.3%, which corresponds to 0.013×(36.6 m/s) = 0.5m/s. Then the discharge velocity can be expressed as

$$V_2 = \mathbf{36.6 \pm 0.5\ m/s}$$

Discussion The error analysis does not include the effects of friction in the duct; the error due to frictional losses is most likely more severe than the error calculated here.

5-106

Solution A tap is opened on the wall of a very large tank that contains air. The maximum flow rate of air through the tap is to be determined, and the effect of a larger diameter lead section is to be assessed.

Assumptions Flow through the tap is steady, incompressible, and irrotational with negligible friction (so that the flow rate is maximum, and the Bernoulli equation is applicable).

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K.

Analysis The density of air in the tank is

$$\rho_{air} = \frac{P}{RT} = \frac{102\text{ kPa}}{(0.287\text{ kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\text{ K})} = 1.21\text{ kg/m}^3$$

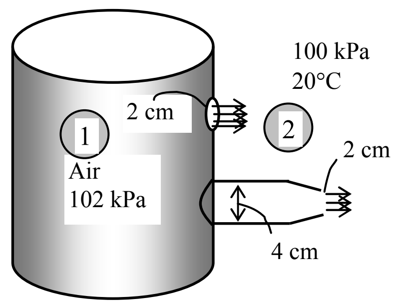
We take point 1 in the tank, and point 2 at the exit of the tap along the same horizontal line. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases) and $V_1 \cong 0$, the Bernoulli equation between points 1 and 2 gives

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

Substituting, the discharge velocity and the flow rate becomes

$$V_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho_{air}}} = \sqrt{\frac{2(102-100)\text{ kN/m}^2}{1.21\text{ kg/m}^3}\left(\frac{1000\text{ kg}\cdot\text{m/s}^2}{1\text{ kN}}\right)} = 57.5\text{ m/s}$$

$$\dot{V} = AV_2 = \frac{\pi D_2^2}{4}V_2 = \frac{\pi(0.02\text{ m})^2}{4}(57.5\text{ m/s}) = \mathbf{0.0181\ m^3/s}$$



This is the *maximum* flow rate since it is determined by assuming frictionless flow. The actual flow rate will be less.

Adding a 2-m long larger diameter lead section will have **no effect** on the flow rate since the flow is frictionless (by using the Bernoulli equation, it can be shown that the velocity in this section increases, but the pressure decreases, and there is a smaller pressure difference to drive the flow through the tab, with zero net effect on the discharge rate).

Discussion If the pressure in the tank were 300 kPa, the flow is no longer incompressible, and thus the problem in that case should be analyzed using compressible flow theory.

5-107

Solution Water is flowing through a venturi meter with known diameters and measured pressures. The flow rate of water is to be determined for the case of frictionless flow.

Assumptions **1** The flow through the venturi is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** The flow is horizontal so that elevation along the centerline is constant. **3** The pressure is uniform at a given cross-section of the venturi meter (or the elevation effects on pressure measurement are negligible).

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

Analysis We take point 1 at the main flow section and point 2 at the throat along the centerline of the venturi meter. Noting that $z_1 = z_2$, the application of the Bernoulli equation between points 1 and 2 gives

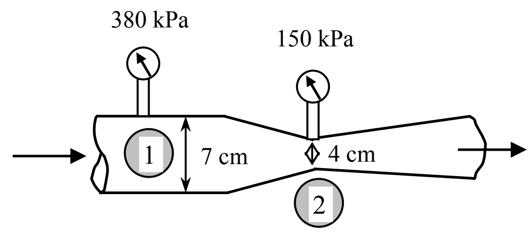
$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 \quad \rightarrow \quad P_1 - P_2 = \rho \frac{V_2^2 - V_1^2}{2} \qquad (1)$$

The flow is assumed to be incompressible and thus the density is constant. Then the conservation of mass relation for this single stream steady flow device can be expressed as

$$\dot{V}_1 = \dot{V}_2 = \dot{V} \quad \rightarrow \quad A_1 V_1 = A_2 V_2 = \dot{V} \quad \rightarrow \quad V_1 = \frac{\dot{V}}{A_1} \quad \text{and} \quad V_2 = \frac{\dot{V}}{A_2} \qquad (2)$$

Substituting into Eq. (1),

$$P_1 - P_2 = \rho \frac{(\dot{V}/A_2)^2 - (\dot{V}/A_1)^2}{2} = \frac{\rho \dot{V}^2}{2A_2^2}\left(1 - \frac{A_2^2}{A_1^2}\right)$$



Solving for $\dot{V}$ gives the desired relation for the flow rate,

$$\dot{V} = A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho[1 - (A_2/A_1)^2]}} \qquad (3)$$

The flow rate for the given case can be determined by substituting the given values into this relation to be

$$\dot{V} = \frac{\pi D_2^2}{4}\sqrt{\frac{2(P_1 - P_2)}{\rho[1 - (D_2/D_1)^4]}} = \frac{\pi (0.04\ \text{m})^2}{4}\sqrt{\frac{2(380 - 150)\ \text{kN/m}^2}{(1000\ \text{kg/m}^3)[1 - (4/7)^4]}\left(\frac{1000\ \text{kg}\cdot\text{m/s}^2}{1\ \text{kN}}\right)} = \mathbf{0.0285\ m^3/s}$$

Discussion Venturi meters are commonly used as flow meters to measure the flow rate of gases and liquids by simply measuring the pressure difference P_1 - P_2 by a manometer or pressure transducers. The actual flow rate will be less than the value obtained from Eq. (3) because of the friction losses along the wall surfaces in actual flow. But this difference can be as little as 1% in a well-designed venturi meter. The effects of deviation from the idealized Bernoulli flow can be accounted for by expressing Eq. (3) as

$$\dot{V} = C_d A_2 \sqrt{\frac{2(P_1 - P_2)}{\rho[1 - (A_2/A_1)^2]}}$$

where C_d is the *venturi discharge coefficient* whose value is less than 1 (it is as large as 0.99 for well-designed venturi meters in certain ranges of flow). For Re > 10^5, the value of venturi discharge coefficient is usually greater than 0.96.

5-108

Solution Water flows through the enlargement section of a horizontal pipe at a specified rate. For a given head loss, the pressure change across the enlargement section is to be determined.

Assumptions **1** The flow through the pipe is steady and incompressible. **2** The pipe is horizontal. **3** The kinetic energy correction factors are given to be $\alpha_1 = \alpha_2 = \alpha = 1.05$.

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

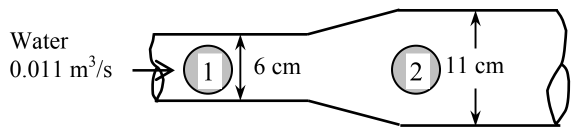
Analysis We take points 1 and 2 at the inlet and exit of the enlargement section along the centerline of the pipe. Noting that $z_1 = z_2$, the energy equation for a control volume between these two points (in terms of heads) simplifies to

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

where the inlet and exit velocities are

$$V_1 = \frac{\dot{V}}{A_1} = \frac{\dot{V}}{\pi D_1^2/4} = \frac{0.011\text{ m}^3/\text{s}}{\pi(0.06\text{ m})^2/4} = 3.890\text{ m/s}$$

$$V_2 = \frac{\dot{V}}{A_2} = \frac{\dot{V}}{\pi D_2^2/4} = \frac{0.011\text{ m}^3/\text{s}}{\pi(0.11\text{ m})^2/4} = 1.157\text{ m/s}$$



Substituting, the change in static pressure across the enlargement section is determined to be

$$P_2 - P_1 = (1000\text{ kg/m}^3)\left(\frac{1.05[(3.890\text{ m/s})^2 - (1.157\text{ m/s})^2]}{2} - (9.81\text{ m/s}^2)(0.65\text{ m})\right)\left(\frac{1\text{ N}}{1\text{ kg}\cdot\text{m/s}^2}\right)\left(\frac{1\text{ kPa}}{1000\text{ N/m}^2}\right)$$

$$= \mathbf{0.865\text{ kPa}}$$

Therefore, the water pressure increases by 0.865 kPa across the enlargement section.

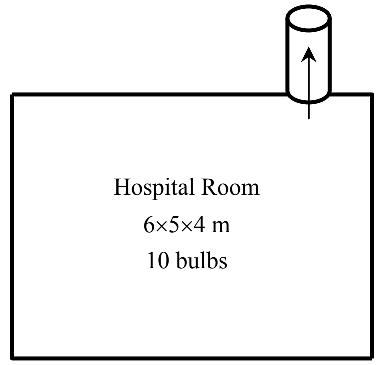
Discussion Note that the pressure increases despite the head loss in the enlargement section. This is due to dynamic pressure being converted to static pressure. But the total pressure (static + dynamic) decreases by 0.65 m (or 6.38 kPa) as a result of frictional effects.

5-109

Solution The air in a hospital room is to be replaced every 20 minutes. The minimum diameter of the duct is to be determined if the air velocity is not to exceed a certain value.

Assumptions **1** The volume occupied by the furniture etc in the room is negligible. **2** The incoming conditioned air does not mix with the air in the room.

Analysis The volume of the room is



$$V = (6\text{ m})(5\text{ m})(4\text{ m}) = 120\text{ m}^3$$

To empty this air in 20 min, the volume flow rate must be

$$\dot{V} = \frac{V}{\Delta t} = \frac{120\text{ m}^3}{20\times 60\text{ s}} = 0.10\text{ m}^3/\text{s}$$

If the mean velocity is 5 m/s, the diameter of the duct is

$$\dot{V} = AV = \frac{\pi D^2}{4}V \quad\rightarrow\quad D = \sqrt{\frac{4\dot{V}}{\pi V}} = \sqrt{\frac{4(0.10\text{ m}^3/\text{s})}{\pi(5\text{ m/s})}} = \mathbf{0.16\ m}$$

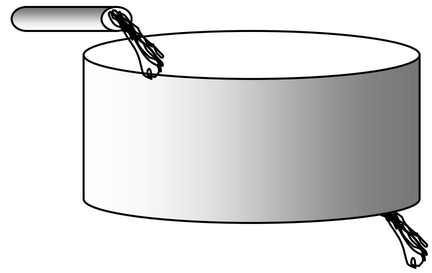
Therefore, the diameter of the duct must be at least 0.16 m to ensure that the air in the room is exchanged completely within 20 min while the mean velocity does not exceed 5 m/s.

Discussion This problem shows that engineering systems are sized to satisfy certain constraints imposed by certain considerations.

5-110

Solution The rate of accumulation of water in a pool and the rate of discharge are given. The rate supply of water to the pool is to be determined.

Assumptions **1** Water is supplied and discharged steadily. **2** The rate of evaporation of water is negligible. **3** No water is supplied or removed through other means.



Analysis The conservation of mass principle applied to the pool requires that the rate of increase in the amount of water in the pool be equal to the difference between the rate of supply of water and the rate of discharge. That is,

$$\frac{dm_{\text{pool}}}{dt} = \dot{m}_i - \dot{m}_e \quad \rightarrow \quad \dot{m}_i = \frac{dm_{\text{pool}}}{dt} + \dot{m}_e \quad \rightarrow \quad \dot{V}_i = \frac{d\dot{V}_{\text{pool}}}{dt} + \dot{V}_e$$

since the density of water is constant and thus the conservation of mass is equivalent to conservation of volume. The rate of discharge of water is

$$\dot{V}_e = A_e V_e = (\pi D^2/4)V_e = [\pi(0.05\text{ m})^2/4](5\text{ m/s}) = 0.00982\text{ m}^3/\text{s}$$

The rate of accumulation of water in the pool is equal to the cross-section of the pool times the rate at which the water level rises,

$$\frac{dV_{\text{pool}}}{dt} = A_{\text{cross-section}} V_{\text{level}} = (3\text{ m} \times 4\text{ m})(0.015\text{ m/min}) = 0.18\text{ m}^3/\text{min} = 0.00300\text{ m}^3/\text{s}$$

Substituting, the rate at which water is supplied to the pool is determined to be

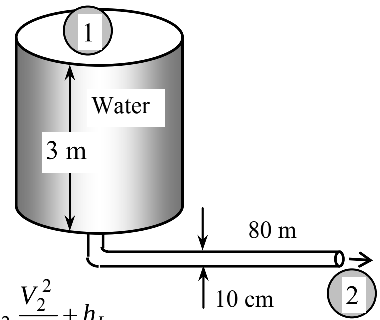
$$\dot{V}_i = \frac{dV_{\text{pool}}}{dt} + \dot{V}_e = 0.003 + 0.00982 = 0.01282\text{ m}^3/\text{s} \cong \mathbf{0.0128\text{ m}^3/s}$$

Therefore, water is supplied at a rate of 0.01282 m^3/s = 12.82 L/s.

Discussion This is a very simple application of the conservation of mass equations.

5-111

Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere through a long pipe with a specified head loss. The initial discharge velocity is to be determined.



Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** There are no pumps or turbines in the system. **4** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the pipe. We take the reference level at the centerline of the orifice (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 very low ($V_1 \cong 0$), the energy equation 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 $\alpha_2 = 1$ and the head loss is given to be $h_L = 1.5$ m. Solving for V_2 and substituting, the discharge velocity of water is determined to be

$$V_2 = \sqrt{2g(z_1 - h_L)} = \sqrt{2(9.81\ \text{m/s}^2)(3-1.5)\ \text{m}} = \mathbf{5.42\ \ m/s}$$

Discussion Note that this is the discharge velocity at the beginning, and the velocity will decrease as the water level in the tank drops. The head loss in that case will change since it depends on velocity.

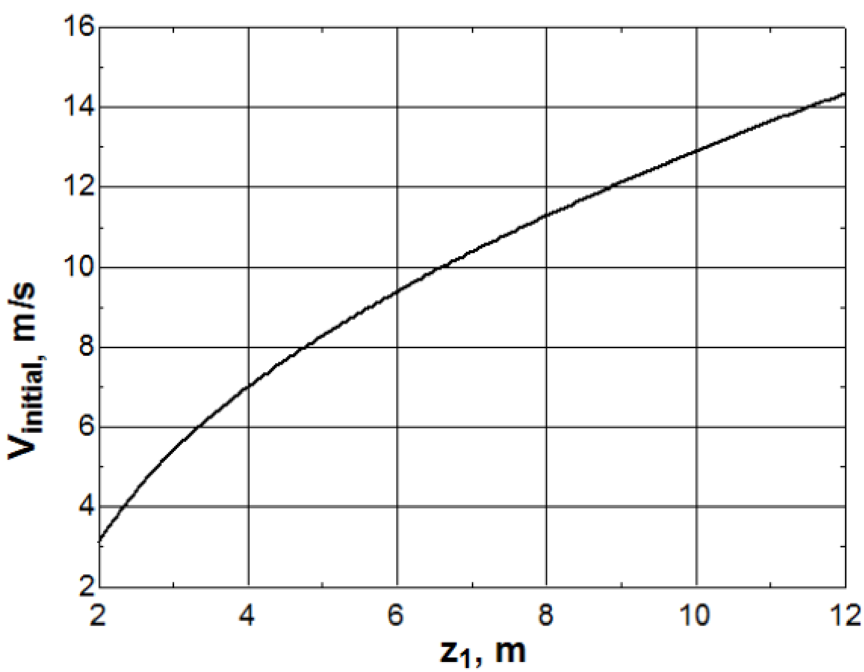
5-112

Solution The previous problem is reconsidered. The effect of the tank height on the initial discharge velocity of water from the completely filled tank as the tank height varies from 2 to 20 m in increments of 2 m at constant heat loss is to be investigated.

Analysis The EES *Equations* window is printed below, followed by the tabulated and plotted results.

```
g=9.81 "m/s2"
rho=1000 "kg/m3"
h_L=1.5 "m"
D=0.10 "m"
V_initial=SQRT(2*g*(z1-h_L)) "m/s"
```

Tank height, z1, m	Head Loss, h_L, m	Initial velocity $V_{initial}$, m/s
2	1.5	3.13
3	1.5	5.42
4	1.5	7.00
5	1.5	8.29
6	1.5	9.40
7	1.5	10.39
8	1.5	11.29
9	1.5	12.13
10	1.5	12.91
11	1.5	13.65
12	1.5	14.35



Discussion The dependence of V on height is not linear, but rather V changes as the square root of z_1.

5-113

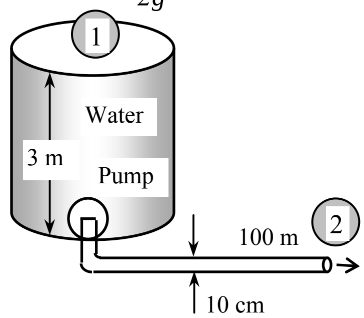
Solution A water tank open to the atmosphere is initially filled with water. A sharp-edged orifice at the bottom drains to the atmosphere through a long pipe equipped with a pump with a specified head loss. The required pump head to assure a certain velocity is to be determined.

Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The effect of the kinetic energy correction factor is negligible, $\alpha = 1$.

Analysis We take point 1 at the free surface of the tank, and point 2 at the exit of the pipe. We take the reference level at the centerline of the orifice ($z_2 = 0$), and take the positive direction of z to be upwards. 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 very low ($V_1 \cong 0$), the energy equation between these two points (in terms of heads) simplifies to

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

where $\alpha_2 = 1$ and the head loss is given to be $h_L = 1.5$ m. Solving for $h_{\text{pump, u}}$ and substituting, the required useful pump head is determined to be



$$h_{pum\,,u} = \sqrt{\frac{V_2^2}{2g} - z_1 + h_L} = \sqrt{\frac{(6.5\text{ m/s})^2}{2(9.81\text{ m/s}^2)} - (3\text{ m}) + (1.5\text{ m})} = \mathbf{0.808\ m}$$

Discussion Note that this is the required useful pump head at the beginning, and it will need to be increased as the water level in the tank drops to make up for the lost elevation head to maintain the constant discharge velocity.

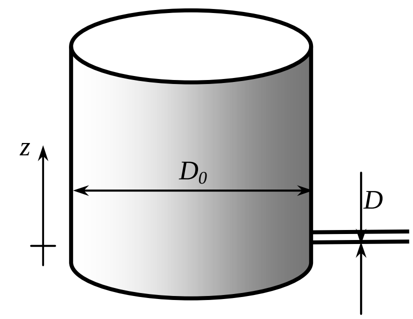
5-114

Solution A water tank open to the atmosphere is initially filled with water. The tank discharges to the atmosphere through a long pipe connected to a valve. The initial discharge velocity from the tank and the time required to empty the tank are to be determined.

Assumptions **1** The flow is incompressible. **2** The draining pipe is horizontal. **3** The tank is considered to be empty when the water level drops to the center of the valve.

Analysis (*a*) Substituting the known quantities, the discharge velocity can be expressed as

$$V = \sqrt{\frac{2gz}{1.5 + fL/D}} = \sqrt{\frac{2gz}{1.5 + 0.015(80\text{ m})/(0.10\text{ m})}} = \sqrt{0.1481gz}$$



Then the initial discharge velocity becomes

$$V_1 = \sqrt{0.1481gz_1} = \sqrt{0.1481(9.81\text{ m/s}^2)(2\text{ m})} = \mathbf{1.705\text{ m/s}}$$

where z is the water height relative to the center of the orifice at that time.

(*b*) The flow rate of water from the tank can be obtained by multiplying the discharge velocity by the pipe cross-sectional area,

$$\dot{V} = A_{\text{pipe}}V_2 = \frac{\pi D^2}{4}\sqrt{0.1481gz}$$

Then the amount of water that flows through the pipe during a differential time interval dt is

$$dV = \dot{V}dt = \frac{\pi D^2}{4}\sqrt{0.1481gz}\,dt \qquad (1)$$

which, from conservation of mass, must be equal to the decrease in the volume of water in the tank,

$$dV = A_{\tan k}(-dz) = -\frac{\pi D_0^2}{4}dz \qquad (2)$$

where dz is the change in the water level in the tank during dt. (Note that dz is a negative quantity since the positive direction of z is upwards. Therefore, we used $-dz$ to get a positive quantity for the amount of water discharged). Setting Eqs. (1) and (2) equal to each other and rearranging,

$$\frac{\pi D^2}{4}\sqrt{0.1481gz}\,dt = -\frac{\pi D_0^2}{4}dz \;\rightarrow\; dt = -\frac{D_0^2}{D^2}\frac{dz}{\sqrt{0.1481gz}} = -\frac{D_0^2}{D^2\sqrt{0.1481g}}z^{-\frac{1}{2}}dz$$

The last relation can be integrated easily since the variables are separated. Letting t_f be the discharge time and integrating it from $t = 0$ when $z = z_1$ to $t = t_f$ when $z = 0$ (completely drained tank) gives

$$\int_{t=0}^{t_f} dt = -\frac{D_0^2}{D^2\sqrt{0.1481g}}\int_{z=z_1}^{0} z^{-1/2}dz \;\rightarrow\; t_f = -\frac{D_0^2}{D^2\sqrt{0.1481g}}\left|\frac{z^{\frac{1}{2}}}{\frac{1}{2}}\right|_{z_1}^{0} = \frac{2D_0^2}{D^2\sqrt{0.1481g}}z_1^{\frac{1}{2}}$$

Simplifying and substituting the values given, the draining time is determined to be

$$t_f = \frac{2D_0^2}{D^2}\sqrt{\frac{z_1}{0.1212g}} = \frac{2(8\text{ m})^2}{(0.1\text{ m})^2}\sqrt{\frac{2\text{ m}}{0.1481(9.81\text{ m/s}^2)}} = 15{,}018\text{ s} = \mathbf{4.17\text{ h}}$$

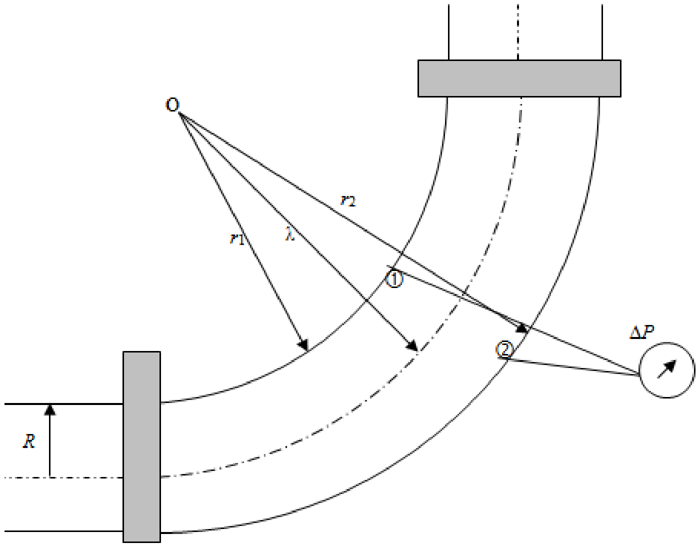
Discussion The draining time can be shortened considerably by installing a pump in the pipe.

5-115

Solution Elbow-type flow meters are used to measure flow rates. A relation for the flow rate as a function of given parameters is to be obtained.

Assumptions **1** The flow is steady and incompressible.

Analysis



Bernoulli equation in the direction r normal to streamlines for steady, incompressible flow is given by (Eq. 5-43)

$$\frac{P}{\rho}+\int\frac{V^2}{r}dr+gz=K \text{ (constant)}$$

where r is the local radius of curvature. Substituting $V=C/r$ and performing integration gives

$$\frac{P}{\rho}+\frac{V^2}{2}+gz=K_1 \text{ (another constant)} \qquad (1)$$

Applying to points 1 and 2 and taking $z_1=z_2$ gives

$$\frac{P_1}{\rho}+\frac{V_1^2}{2}=\frac{P_2}{\rho}+\frac{V_2^2}{2} \text{ or } \frac{\Delta P}{\rho}=\frac{V_1^2-V_2^2}{2} \qquad (1)$$

Substituting $V_1=\dfrac{C}{r_1}$ and $V_2=\dfrac{C}{r_2}$ into Eq. 1 and solving for C,

$$C=\sqrt{2\frac{\Delta p}{\rho}}\frac{r_1 r_2}{\sqrt{r_2^2-r_1^2}} \qquad (2)$$

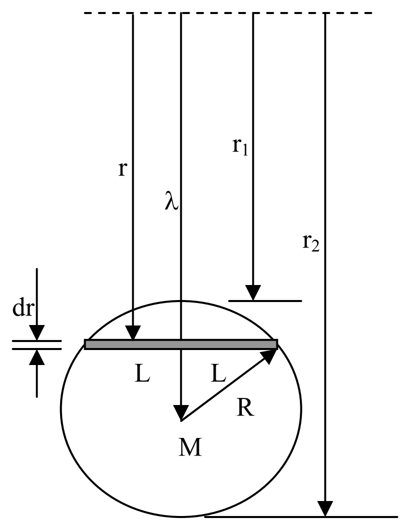
But from the figure, $r_1=\lambda-R$ and $r_2=\lambda+R$. Substituting, Eq. 2 becomes

$$C=(\lambda^2-R^2)\sqrt{\frac{\Delta p}{2\rho\lambda R}} \qquad (3)$$

The flow rate is given by $Q=\int_A VdA$. Let us determine what dA would be. Considering the cross-sectional area of the elbow below we can write

$$dA=(2L)dr$$

From the sketch, $L = \sqrt{R^2 - (\lambda - r)^2}$



$$dA = 2\sqrt{R^2 - (\lambda - r)^2}\,dr$$

$$Q = \int V(2L)dr = 2\int \frac{C}{r}\sqrt{R^2 - (\lambda - r)^2}\,dr$$

$$Q = 2C\int_{r_1=\lambda-R}^{r_2=\lambda+R} \frac{\sqrt{R^2 - (\lambda - r)^2}}{r}dr$$

$$= 2\pi C\left(\lambda - \sqrt{\lambda^2 - R^2}\right)$$

Combining with Eq. 3, we obtain the flow rate to be

$$Q = \pi\sqrt{\frac{2\Delta P}{\rho g\lambda R}}\left(\lambda^2 - R^2\right)\left(\lambda - \sqrt{\lambda^2 - R^2}\right)$$

5-116

Solution A cylindrical water tank with a valve at the bottom contains air at the top part and water. The water height in the tank when water stops flowing out of the fully open valve is to be determined.

Assumptions **1** The flow is incompressible.

Analysis Applying Bernoulli Eq. from free surface to the point A would give

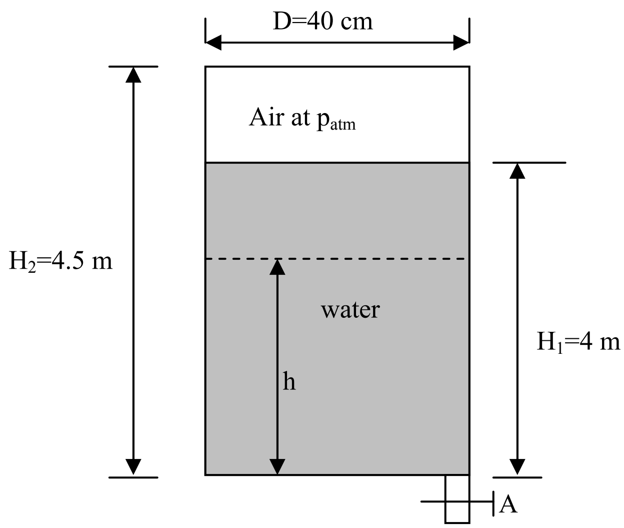
$$\frac{P_{air}}{\gamma}+\frac{V_{air}^{\ 2}}{2g}+Z_{air}=\frac{P_{atm}}{\gamma}+\frac{V_A^{\ 2}}{2g}+Z_A$$

$$P_{air,initial}\mathcal{V}_{air,initial}=P_{air}\mathcal{V}_{air}=mRT=\text{constant}$$

$$P_{atm}\frac{\pi D^2}{4}(H_1-H_2)=P_{air}(H_1-h)$$

$$100000\times(4.5-4)=P_{air}(4.5-h)$$

$$P_{air}=\frac{50000}{4.5-h}$$



Plugging P_{air} into the Bernoulli Equation,

$$\frac{50000}{9810(4.5-h)}+0+h=\frac{10000}{9810}+\frac{V_A^{\ 2}}{2g}+Z_A$$

$$\frac{5.097}{4.5-h}+h=10.193+\frac{V_A^{\ 2}}{2g}$$

In case of no flow, V_A=0, that is

$$5.097+4.5h-h^2=45.87-10.193h$$

$$h^2-14.693h+40.773=0$$

$$h_1=10.98m \qquad \text{(not possible)}$$

$$h_2=\mathbf{3.71\ m}$$

Therefore, when the water level becomes h=3.71m the flow would stop. The discharged water volume is then

$$V=\pi\frac{D^2}{4}(H_2-h)=\pi\frac{0.4^2}{4}(4-3.71)=0.0364\ m^3$$

5-117

Solution A compressor supplies air to a rigid tank. A relation for the variation of pressure in the tank with time is to be obtained and and the time it will take for the absolute pressure in the tank to triple is to be determined.

Analysis (a) Applying conservation of mass to the CV enclosing tank, we get

$$\frac{\partial}{\partial t}\int_{CV}\rho d\forall + \int_{CS}\rho\vec{V}\vec{n}dA = 0\,, \text{ and } \forall\frac{d\rho}{dt} - \rho Q = 0$$

Therefore, we write

$$\int_{\rho_0}^{\rho}\frac{d\rho}{\rho} = \frac{Q}{\forall}\int_0^t dt \qquad \ln\frac{\rho}{\rho_0} = \frac{Q}{\forall}t$$

$$\frac{\rho}{\rho_0} = e^{\frac{Q}{\forall}t} \tag{1}$$

On the other hand, considering an adiabatic flow we have,

$$\frac{P}{\rho^k} = \frac{P_0}{\rho_0{}^k} \Rightarrow \left(\frac{\rho}{\rho_0}\right)^k = \frac{P}{P_0} \tag{2}$$

From the Equation 1;

$$\left(\frac{\rho}{\rho_0}\right)^k = e^{\frac{Q}{\forall}kt} = \frac{P}{P_0}$$

Therefore

$$P = P_0 e^{\frac{Q}{\forall}kt}$$

(b) $P = 3P_0$

$$3P_0 = P_0 e^{\frac{Q}{\forall}kt}\text{, or } \ln 3 = \frac{Q}{\forall}kt$$

$$t = \frac{\forall}{Q}.\frac{\ln 3}{k} = \frac{1.5}{0.05}.\frac{\ln 3}{1.4} = \mathbf{23.5\ s}$$

5-118

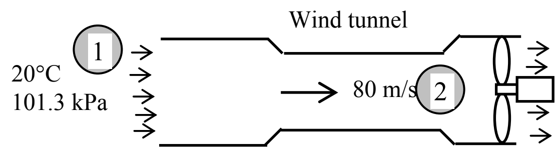
Solution A wind tunnel draws atmospheric air by a large fan. For a given air velocity, the pressure in the tunnel is to be determined.

Assumptions **1**The flow through the pipe is steady, incompressible, and irrotational with negligible friction (so that the Bernoulli equation is applicable). **2** Air is and ideal gas.

Properties The gas constant of air is $R = 0.287$ kPa·m^3/kg·K.

Analysis We take point 1 in atmospheric air before it enters the wind tunnel (and thus $P_1 = P_{atm}$ and $V_1 \cong 0$), and point 2 in the wind tunnel. Noting that $z_1 = z_2$ (or, the elevation effects are negligible for gases), the Bernoulli equation between points 1 and 2 gives

$$\frac{P_1}{\rho g}+\frac{V_1^2}{2g}+z_1=\frac{P_2}{\rho g}+\frac{V_2^2}{2g}+z_2 \quad \rightarrow \quad P_2 = P_1 - \frac{\rho V_2^2}{2} \qquad (1)$$



where

$$\rho = \frac{P}{RT} = \frac{101.3\ \text{kPa}}{(0.287\ \text{kPa}\cdot\text{m}^3/\text{kg}\cdot\text{K})(293\ \text{K})} = 1.205\ \text{kg/m}^3$$

Substituting, the pressure in the wind tunnel is determined to be

$$P_2 = (101.3\ \text{kPa}) - (1.205\ \text{kg/m}^3)\frac{(80\ \text{m/s})^2}{2}\left(\frac{1\ \text{N}}{1\ \text{kg}\cdot\text{m/s}^2}\right)\left(\frac{1\ \text{kPa}}{1000\ \text{N/m}^2}\right) = \mathbf{97.4\ kPa}$$

Discussion Note that the velocity in a wind tunnel increases at the expense of pressure. In reality, the pressure will be even lower because of losses.

Fundamentals of Engineering (FE) Exam Problems

5-119

Water flows in a 5-cm-diameter pipe at a velocity of 0.75 m/s. The mass flow rate of water in the pipe is

(*a*) 353 kg/min (*b*) 75 kg/min (*c*) 37.5 kg/min (*d*) 1.47 kg/min (*e*) 88.4 kg/min

Answer (*e*) 88.4 kg/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
D=0.05 [m]
V=0.75 [m/s]
rho=1000 [kg/m^3]
A_c=pi*D^2/4
m_dot=rho*A_c*V*Convert(kg/s, kg/min)
```

5-120

Air at 100 kPa and 20°C flows in a 12-cm-diameter pipe at a rate of 9.5 kg/min. The velocity of air in the pipe is

(*a*) 1.4 m/s (*b*) 6.0 m/s (*c*) 9.5 m/s (*d*) 11.8 m/s (*e*) 14.0 m/s

Answer (*d*) 11.8 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P=100 [kPa]
T=(20+273.15) [K]
D=0.12 [m]
m_dot=9.5 [kg/min]*Convert(kg/min, kg/s)
R=0.287 [kJ/kg-K]
rho=P/(R*T)
A_c=pi*D^2/4
V=m_dot/(rho*A_c)
```

5-121

A water tank initially contains 140 L of water. Now, equal rates of cold and hot water enter the tank for a period of 30 minutes while warm water is discharged from the tank at a rate of 25 L/min. The amount of water in the tank at the end of this 30-min period is 50 L. The rate of hot water entering the tank is

(*a*) 33 L/min (*b*) 25 L/min (*c*) 11 L/min (*d*) 7 L/min (*e*) 5 L/min

Answer (*c*) 11 L/min

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

V1=140 [L]

time=30 [min]

V_dot_out=25 [L/min]

V2=50 [L]

V_out=V_dot_out*time

V_in-V_out=V2-V1

V_in=V_cold+V_hot

V_cold=V_hot

V_dot_hot=V_hot/time

5-122

Water enters a 4-cm-diameter pipe at a velocity of 1 m/s. The diameter of the pipe is reduced to 3 cm at the exit. The velocity of the water at the exit is

(*a*) 1.78 m/s (*b*) 1.25 m/s (*c*) 1 m/s(*d*) 0.75 m/s (*e*) 0.50 m/s

Answer (*a*) 1.78 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

D1=0.04 [m]

V1=1 [m/s]

D2=0.03 [m]

A_c1=pi*D1^2/4

A_c2=pi*D2^2/4

A_c1*V1=A_c2*V2

5-123

The pressure of water is increased from 100 kPa to 900 kPa by a pump. The mechanical energy increase of water is

(*a*) 0.9 kJ/kg (*b*) 0.5 kJ/kg (*c*) 500 kJ/kg (*d*) 0.8 kJ/kg (*e*) 800 kJ/kg

Answer (*d*) 0.8 kJ/kg

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

P1=100 [kPa]

P2=900 [kPa]

rho=1000 [kg/m^3]

DELTAe=(P2-P1)/rho

5-124

A 75-m-high water body that is open to the atmosphere is available. Water is run through a turbine at a rate of 200 L/s at the bottom of the water body. The pressure difference across the turbine is

(*a*) 736 kPa (*b*) 0.736 kPa (*c*) 1.47 kPa (*d*) 1470 kPa (*e*) 368 kPa

Answer (*a*) 736 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

h=75 [m]

g=9.81 [m/s^2]

rho=1000 [kg/m^3]

e=g*h*Convert(J/kg, kJ/kg)

DELTAP=e*rho

5-125

A pump is used to increase the pressure of water from 100 kPa to 900 kPa at a rate of 160 L/min. If the shaft power input to the pump is 3 kW, the efficiency of the pump is

(*a*) 0.532 (*b*) 0.660 (*c*) 0.711 (*d*) 0.747 (*e*) 0.855

Answer (*c*) 0.711

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=900 [kPa]
V_dot=160 [L/min]*Convert(L/min, m^3/s)
W_dot_shaft=3 [kW]
rho=1000 [kg/m^3]
m_dot=rho*V_dot
DELTAE_dot=m_dot*(P2-P1)/rho
eta_pump=DELTAE_dot/W_dot_shaft
```

5-126

A hydraulic turbine is used to generate power by using the water in a dam. The elevation difference between the free surfaces upstream and downstream of the dam is 120 m. The water is supplied to the turbine at a rate of 150 kg/s. If the shaft power output from the turbine is 155 kW, the efficiency of the turbine is

(*a*) 0.77 (*b*) 0.80 (*c*) 0.82 (*d*) 0.85 (*e*) 0.88

Answer (*e*) 0.88

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h=120 [m]
m_dot=150 [kg/s]
W_dot_shaft=155 [kW]
g=9.81 [m/s^2]
DELTAE_dot=m_dot*g*h*Convert(W, kW)
eta_turbine=W_dot_shaft/DELTAE_dot
```

5-127

The motor of a pump consumes 1.05 hp of electricity. The pump increases the pressure of water from 120 kPa to 1100 kPa at a rate of 35 L/min. If the motor efficiency is 94 percent, the pump efficiency is

(*a*) 0.75 (*b*) 0.78 (*c*) 0.82 (*d*) 0.85 (*e*) 0.88

Answer (*b*) 0.78

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

W_dot_elec=1.05 [hp]*Convert(hp, kW)

P1=120 [kPa]

P2=1100 [kPa]

V_dot=35 [L/min]*Convert(L/min, m^3/s)

eta_motor=0.94

rho=1000 [kg/m^3]

m_dot=rho*V_dot

DELTAE_dot=m_dot*(P2-P1)/rho

eta_motor_pump=DELTAE_dot/W_dot_elec

eta_pump=eta_motor_pump/eta_motor

5-128

The efficiency of a hydraulic turbine-generator unit is specified to be 85 percent. If the generator efficiency is 96 percent, the turbine efficiency is

(*a*) 0.816 (*b*) 0.850 (*c*) 0.862 (*d*) 0.885 (*e*) 0.960

Answer (*d*) 0.885

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

eta_turbine_gen=0.85

eta_gen=0.96

eta_turbine=eta_turbine_gen/eta_gen

5-129

Which parameter is not related in the Bernoulli equation?

(*a*) Density (*b*) Velocity (*c*) Time (*d*) Pressure (*e*) Elevation

Answer (*c*) Time

5-130

Consider incompressible, frictionless flow of a fluid in a horizontal piping. The pressure and velocity of a fluid is measured to be 150 kPa and 1.25 m/s at a specified point. The density of the fluid is 700 kg/m^3. If the pressure is 140 kPa at another point, the velocity of the fluid at that point is

(*a*) 1.26 m/s (*b*) 1.34 m/s (*c*) 3.75 m/s (*d*) 5.49 m/s (*e*) 7.30 m/s

Answer (*d*) 5.49 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=150 [kPa]
V1=1.25 [m/s]
rho=700 [kg/m^3]
P2=140 [kPa]
P1/rho+V1^2/2*Convert(m^2/s^2, kJ/kg)=P2/rho+V2^2/2*Convert(m^2/s^2, kJ/kg)
```

5-131

Consider incompressible, frictionless flow of water in a vertical piping. The pressure is 240 kPa at 2 m from the ground level. The velocity of water does not change during this flow. The pressure at 15 m from the ground level is

(*a*) 227 kPa (*b*) 174 kPa (*c*) 127 kPa (*d*) 120 kPa (*e*) 113 kPa

Answer (*e*) 113 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=240 [kPa]
z1=2 [m]
z2=15 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P1/rho+g*z1*Convert(m^2/s^2, kJ/kg)=P2/rho+g*z2*Convert(m^2/s^2, kJ/kg)
```

5-132

Consider water flow in a piping network. The pressure, velocity, and elevation at a specified point (point 1) of the flow are 150 kPa, 1.8 m/s, and 14 m. The pressure and velocity at point 2 are 165 kPa and 2.4 m/s. Neglecting frictional effects, the elevation at point 2 is

(*a*) 12.4 m (*b*) 9.3 m (*c*) 14.2 m (*d*) 10.3 m (*e*) 7.6 m

Answer (*a*) 12.4 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=150 [kPa]
V1=1.8 [m/s]
z1=14 [m]
P2=165 [kPa]
V2=2.4 [m/s]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P1/rho+(V1^2/2+g*z1)*Convert(m^2/s^2, kJ/kg)=P2/rho+(V2^2/2+g*z2)*Convert(m^2/s^2, kJ/kg)
```

5-133

The static and stagnation pressures of a fluid in a pipe are measured by a piezometer and a pitot tube to be 200 kPa and 210 kPa, respectively. If the density of the fluid is 550 kg/m^3, the velocity of the fluid is

(*a*) 10 m/s (*b*) 6.03 m/s (*c*) 5.55 m/s (*d*) 3.67 m/s (*e*) 0.19 m/s

Answer (*b*) 6.03 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P_stag=210 [kPa]
P=200 [kPa]
rho=550 [kg/m^3]
P_stag=P+rho*V^2/2*Convert(Pa, kPa)
```

5-134

The static and stagnation pressures of a fluid in a pipe are measured by a piezometer and a pitot tube. The heights of the fluid in the piozemeter and pitot tube are measured to be 2.2 m and 2.0 m, respectively. If the density of the fluid is 5000 kg/m^3, the velocity of the fluid in the pipe is

(*a*) 0.92 m/s (*b*) 1.43 m/s (*c*) 1.65 m/s (*d*) 1.98 m/s (*e*) 2.39 m/s

Answer (*d*) 1.98 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
h_piezo=2 [m]
h_pitot=2.2 [m]
rho=5000 [kg/m^3]
g=9.81 [m/s^2]
P=rho*g*h_piezo
P_stag=rho*g*h_pitot
P_stag=P+rho*V^2/2
```

5-135

The difference between the heights of energy grade line (EGL) and hydraulic grade line (HGL) is equal to

(*a*) z (*b*) $P/\rho g$ (*c*) $V^2/2g$(*d*) $z + P/\rho g$ (*e*) $z + V^2/2g$

Answer (*c*) $V^2/2g$

5-136

Water at 120 kPa (gage) is flowing in a horizontal pipe at a velocity of 1.15 m/s. The pipe makes a 90° angle at the exit and the water exits the pipe vertically into the air. The maximum height the water jet can rise is

(*a*) 6.9 m (*b*) 7.8 m (*c*) 9.4 m (*d*) 11.5 m (*e*) 12.3 m

Answer (*e*) 12.3 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=120000 [Pa]
V1=1.15 [m/s]
z1=0 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P2=0 [Pa]
V2=0 [m/s]
P1/(rho*g)+V1^2/(2*g)+z1=P2/(rho*g)+V2^2/(2*g)+z2
```

5-137

Water is withdrawn at the bottom of a large tank open to the atmosphere. The water velocity is 6.6 m/s. The minimum height of the water in the tank is

(*a*) 2.22 m (*b*) 3.04 m (*c*) 4.33 m (*d*) 5.75 m (*e*) 6.60 m

Answer (*a*) 2.22 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
V2=6.6 [m/s]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P1=0 [Pa]
P2=0 [Pa]
V1=0 [m/s]
z2=0 [m]
P1/(rho*g)+V1^2/(2*g)+z1=P2/(rho*g)+V2^2/(2*g)+z2
```

5-138

Water at 80 kPa (gage) enters a horizontal pipe at a velocity of 1.7 m/s. The pipe makes a 90° angle at the exit and the water exits the pipe vertically into the air. Take the correction factor to be 1. If the irreversible head loss between the inlet and exit of the pipe is 3 m, the height the water jet can rise is

(*a*) 3.4 m (*b*) 5.3 m (*c*) 8.2 m (*d*) 10.5 m (*e*) 12.3 m

Answer (*b*) 5.3 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=80000 [Pa]
V1=1.7 [m/s]
h_L=3 [m]
z1=0 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
P2=0 [Pa]
V2=0 [m/s]
P1/(rho*g)+V1^2/(2*g)+z1=P2/(rho*g)+V2^2/(2*g)+z2+h_L
```

5-139

Seawater is to be pumped into a large tank at a rate of 165 kg/min. The tank is open to the atmosphere and the water enters the tank from a 80-m-height. The overall efficiency of the motor-pump unit is 75 percent and the motor consumes electricity at a rate of 3.2 kW. Take the correction factor to be 1. If the irreversible head loss in the piping is 7 m, the velocity of the water at the tank inlet is

(*a*) 2.34 m/s (*b*) 4.05 m/s (*c*) 6.21 m/s (*d*) 8.33 m/s (*e*) 10.7 m/s

Answer (*c*) 6.21 m/s

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=(165/60) [kg/s]
z2=80 [m]
eta_motor_pump=0.75
W_dot_elec=3200 [W]
h_L=7 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
z1=0 [m]
V1=0 [m/s]
P1=0 [Pa]
P2=0 [Pa]
h_pump_u=eta_motor_pump*W_dot_elec/(m_dot*g)
P1/(rho*g)+V1^2/(2*g)+z1+h_pump_u=P2/(rho*g)+V2^2/(2*g)+z2+h_L
```

5-140

Water enters a pump at 350 kPa at a rate of 1 kg/s. The water leaving the pump enters a turbine in which the pressure is reduced and electricity is produced. The shaft power input to the pump is 1 kW and the shaft power output from the turbine is 1 kW. Bothe the pump and turbine are 90 percent efficient. If the elevation and velocity of the water remain constant throughout the flow and the irreversible head loss is 1 m, the pressure of water at the turbine exit is

(*a*) 350 kPa (*b*) 100 kPa (*c*) 173 kPa (*d*) 218 kPa (*e*) 129 kPa

Answer (*e*) 129 kPa

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
m_dot=1 [kg/s]
P1=350000 [Pa]
W_dot_pump=1000 [W]
W_dot_turbine=1000 [W]
eta_pump=0.90
eta_turbine=0.90
h_L=1 [m]
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
z1=0 [m]
z2=0 [m]
V1=1 [m/s]
V2=1 [m/s]
h_pump_u=eta_pump*W_dot_pump/(m_dot*g)
h_turbine_e=W_dot_turbine/(eta_turbine*m_dot*g)
P1/(rho*g)+V1^2/(2*g)+z1+h_pump_u=P2/(rho*g)+V2^2/(2*g)+z2+h_turbine_e+h_L
```

5-141

An adibatic pump is used to increase the pressure of water from 100 kPa to 500 kPa at a rate of 400 L/min. If the efficiency of the pump is 75 percent, the maximum temperature rise of the water across the pump is

(*a*) 0.096°C (*b*) 0.058°C (*c*) 0.035°C (*d*) 1.52°C (*e*) 1.27°C

Answer (*a*) 0.096°C

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
P1=100 [kPa]
P2=500 [kPa]
V_dot=400 [L/min]*Convert(L/min, m^3/s)
eta_pump=0.75
rho=1000 [kg/m^3]
g=9.81 [m/s^2]
c=4.18 [kJ/kg-C]
m_dot=rho*V_dot
DELTAE_dot_mech=m_dot*(P2-P1)/rho
W_dot_pump=DELTAE_dot_mech/eta_pump
E_dot_mech_loss=W_dot_pump-DELTAE_dot_mech
DELTAT=DELTAE_dot_mech/(m_dot*c)
```

5-142

The shaft power from a 90 percent-efficient turbine is 500 kW. If the mass flow rate through the turbine is 575 kg/s, the extracted head removed from the fluid by the turbine is

(*a*) 48.7 m (*b*) 57.5 m (*c*) 147 m (*d*) 139 m (*e*) 98.5 m

Answer (*e*) 98.5 m

Solution Solved by EES Software. Solutions can be verified by copying-and-pasting the following lines on a blank EES screen. (Similar problems and their solutions can be obtained easily by modifying numerical values).

```
eta_turbine=0.90
W_dot_turbine=500000 [W]
m_dot=575 [kg/s]
g=9.81 [m/s^2]
h_turbine_e=W_dot_turbine/(eta_turbine*m_dot*g)
```

Design and Essay Problems

5-143 to 5-147

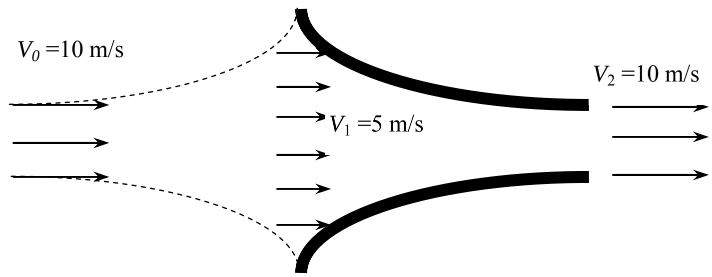
Solution Students' essays and designs should be unique and will differ from each other.

5-148

Solution We are to evaluate a proposed modification to a wind turbine.

Analysis Using the mass and the Bernoulli equations, it can be shown that **this is a bad idea** – the velocity at the exit of nozzle is equal to the wind velocity. (The velocity at nozzle inlet is much lower). Sample calculation using EES using a wind velocity of 10 m/s:

```
V0=10 "m/s"
rho=1.2 "kg/m3"
g=9.81 "m/s2"
A1=2 "m2"
A2=1 "m2"
A1*V1=A2*V2
P1/rho+V1^2/2=V2^2/2
m=rho*A1*V1
m*V0^2/2=m*V2^2/2
```



Results: V_1 = 5 m/s, V_2 = 10 m/s, m = 12 kg/s (mass flow rate).

Discussion Students' approaches may be different, but they should come to the same conclusion – this does not help.