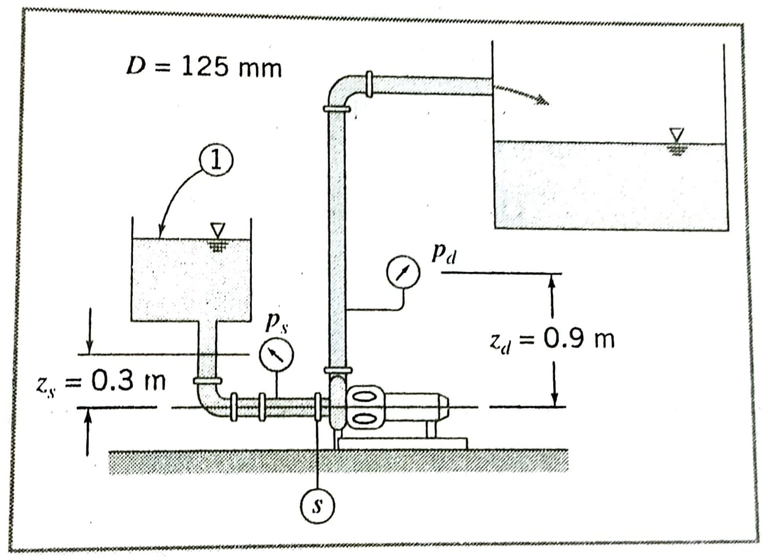
EXAMPLE 10.8 Calculation of Net Positive Suction Head (*NPSH*)

A Peerless Type 4AE11 centrifugal pump (Fig. D.3, Appendix D) is tested at 1750 rpm using a flow system with the layout of Example 10.3. The water level in the inlet reservoir is 1 m above the pump centerline; the inlet line consists of 1.8 m of 125 mm diameter straight cast-iron pipe, a standard elbow, and a fully open gate valve. Calculate the net positive suction head available (*NPSHA*) at the pump inlet at a volume flow rate of 230 m^3/h of water at 30°C. Compare with the net positive suction head required (*NPSHR*) by the pump at this flow rate. Plot *NPSHA* and *NPSHR* for water at 30°C and 80°C versus volume flow rate.

GIVEN: A Peerless Type 4AE11 centrifugal pump (Fig. D.3, Appendix D) is tested at 1750 rpm using a flow system with the layout of Example 10.3. The water level in the inlet reservoir is 1 m above the pump centerline; the inlet line has 1.8 m of 125 mm diameter straight cast-iron pipe, a standard elbow, and a fully open gate valve.



FIND: (a) *NPSHA* at $Q = 230\ m^3/h$ of water at 30°C.
(b) Comparison with *NPSHR* for this pump at $Q = 230\ m^3/h$.
(c) Plot of *NPSHA* and *NPSHR* for water at 30°C and 80°C versus volume flow rate.

SOLUTION:

Net positive suction head (*NPSH*) is defined as the difference between the absolute stagnation pressure in the flow at the pump suction and the liquid vapor pressure, expressed as head of flowing liquid. Therefore it is necessary to calculate the head at the pump suction.

Apply the energy equation for steady, incompressible pipe flow to compute the pressure at the pump inlet and thus the *NPSHA*. Denote the reservoir level as (1) and the pump suction as (s), as shown above.

Governing equation:

$$p_1 + \overset{\approx 0}{\cancel{\frac{1}{2}\rho\overline{V}_1^2}} + \rho g z_1 = p_s + \frac{1}{2}\rho\overline{V}_s^2 + \rho \overset{x}{g_s} + \rho h_{\ell_T}$$

Assumption: $\overline{V}_1$ is negligible. Thus

$$p_s = p_1 + \rho g(z_1 - z_s) - \frac{1}{2}\rho\overline{V}_s^2 - \rho h_{\ell_T}$$

The total head loss is

$$h_{\ell_T} = \left(\sum K + \sum f\frac{L_e}{D} + f\frac{L}{D}\right)\frac{1}{2}\rho\overline{V}_s^2$$

Substituting Eq. 2 into Eq. 1 and dividing by ρg,

$$H_s = H_1 + z_1 - z_s - \left(\sum K + \sum f\frac{L_e}{D} + f\frac{L}{D} + 1\right)\frac{\overline{V}_s^2}{2g}$$

Evaluating the friction factor and head loss,

$$f = f(Re, e/D); \quad Re = \frac{\rho\overline{V}D}{\mu} = \frac{\overline{V}D}{\nu}; \quad \overline{V} = \frac{Q}{A}; \quad A = \frac{\pi D^2}{4}$$

For 125 mm (nominal) pipe, $D = 128$ mm

$$D = 128\text{ mm} \times \frac{\text{m}}{1000\text{ mm}} = 0.128\text{ m}, \quad A = \frac{\pi D^2}{4} = 0.0129\text{ m}^2$$

$$\overline{V} = \frac{230\ \text{m}^3}{\text{h}} \times \frac{\text{h}}{3600\text{s}} \times \frac{1}{0.0129\text{ m}^2} = 4.95\text{ m/s}$$

From Table A.8, for water at $T = 30°\text{C}$, $\nu = 8.03 \times 10^{-7}\text{ m}^2/\text{s}$.

The Reynolds number is

$$Re = \frac{\overline{V}D}{\nu} = \frac{4.95\ \text{m}}{\text{s}} \times 0.128\text{ m} \times \frac{\text{s}}{8.03 \times 10^{-7}\text{ m}^2} = 7.89 \times 10^5$$

From Table 8.1, $e = 0.26$ mm, so $e/D = 0.00203$. From Eq. 8.37, $f = 0.0237$. The minor loss coefficients are

Entrance $K = 0.5$

Standard elbow $\frac{L_e}{D} = 30$

Open gate value $\frac{L_e}{D} = 8$

Substituting,

$$\left(\sum K + \sum f\frac{L_e}{D} + f\frac{L}{D} + 1\right) = 0.5 + 0.0237(30 + 8) + 0.0237\left(\frac{1.8}{0.128}\right) + 1 = 2.73$$

The heads are

$$H_1 = \frac{p_{\text{atm}}}{\rho g} = \frac{1.01325 \times 10^5\text{ Pa}}{996\text{ kg/m}^3 \times 9.8\text{ m/s}^2} \times \frac{\text{N}}{\text{Pa}\cdot\text{m}^2} \times \frac{\text{kg}\cdot\text{m}}{\text{N}\cdot\text{s}^2} = 10.4\text{ m}$$

$$\frac{\overline{V}_s^2}{2g} = \frac{1}{2} \times \frac{(4.95)^2\ \text{m}^2}{\text{s}^2} \times \frac{\text{s}^2}{9.8\text{ m}} = 1.25\text{ m}$$

Thus,

$$H_s = 10.4\text{ m} + 1\text{ m} - 2.73 \times 1.25\text{ m} = 7.98\text{ m}$$

To obtain *NPHSA*, add velocity head and subtract vapor head. Thus

$$NPHSA = H_s + \frac{\overline{V}_s^2}{2g} - H_v$$

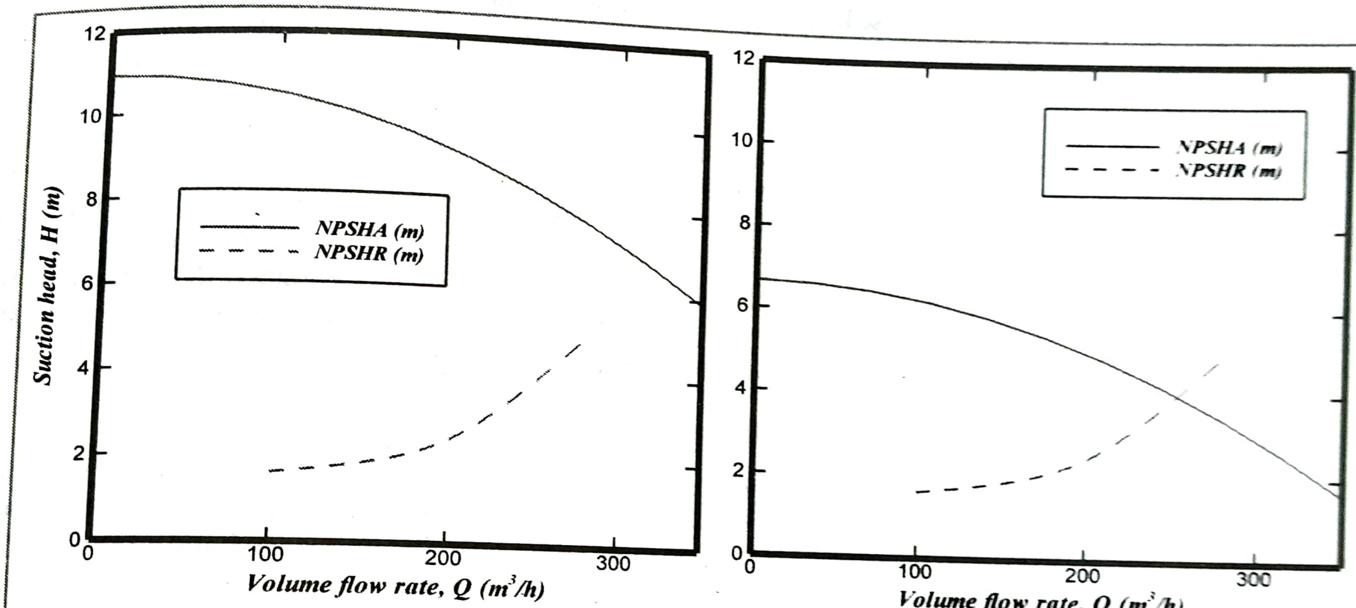
The vapor pressure for water at 30°C is $p_v = 4.25$ kPa. The corresponding head is $H_v = 0.44$ m of water. Thus,

$$NPSHA = 7.98 + 1.25 - 0.44 = 8.79 \text{ m} \qquad \longleftarrow \qquad NPSHA$$

The pump curve (Fig. D.3, Appendix D) shows that at 230 m³/h the pump requires

$$NPSHR = 3.1 \text{ m} \qquad \longleftarrow \qquad NPSHR$$

Results of similar computations for water at 30°C are plotted in the figure on the left below. (*NPSHR* values are obtained from the pump curves in Fig. D.3, Appendix D.)



(*a*) Net positive suction head, water at 30°C

(*b*) Net positive suction head, water at 80°C

Results of computation for water at 80°C are plotted in the figure on the right above. The vapor pressure for water at 80°C is $p_v = 47.4$ kPa. The corresponding head is $H_v = 4.98$ m of water. This high vapor pressure reduces the *NPSHA*, as shown in the plot.

> This problem illustrates the procedures used for checking whether a given pump is in danger of experiencing cavitation:
>
> ✓ Equation 3 and the plots show that the *NPSHA* decreases as flow rate Q (or $\overline{V}_s$) increases; on the other hand, the *NPSHR* increases with Q, so if the flow rate is high enough, a pump will likely experience cavitation (when *NPSHA* < *NPSHR*).
>
> ✓ The *NPSHR* for any pump increases with flow rate Q because local fluid velocities within the pump increase, causing locally reduced pressures and tending to promote cavitation.
>
> ✓ For this pump, at 30°C, the pump appears to have *NPSHA* > *NPSHR* at all flow rates, so it would never experience cavitation; at 80°C, cavitation would occur around 250 m³/h, but from Fig. D.3, the pump best efficiency is around 200 m³/h, so it would probably not be run at 250 m³/h—the pump would probably not cavitate even with the hotter water.
>
> The *Excel* workbook for this Example can be used to generate the *NPSHA* and *NPSHR* curves for a variety of pumps and water temperatures.