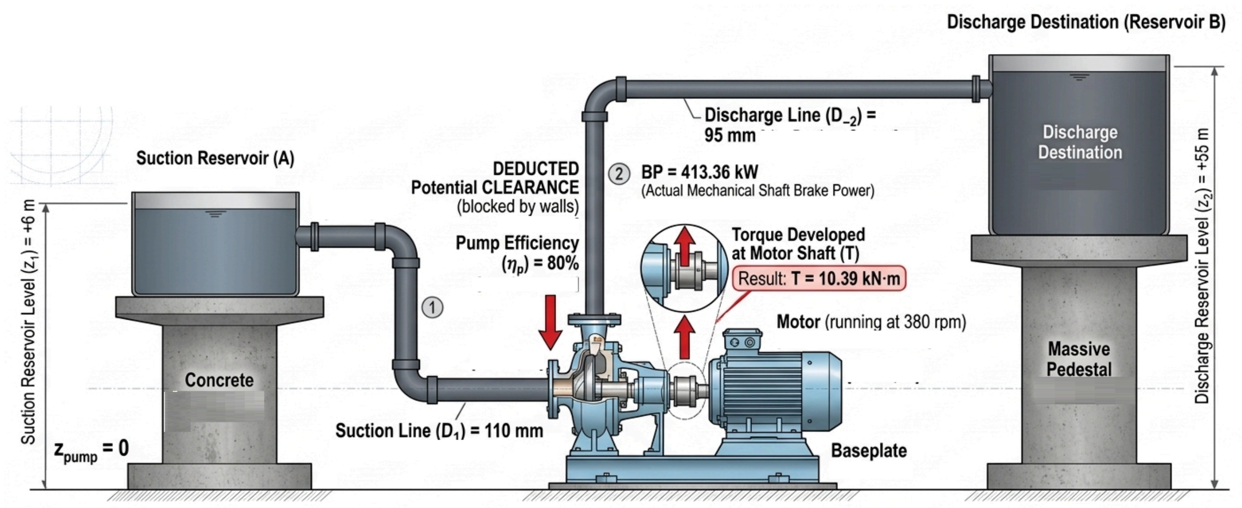
To solve for the mechanical torque developed at the motor shaft of a continuous pumping system, we must conduct a complete energy accounting using the **Total Dynamic Head (TDH)** framework. The system consists of a pump drawing water from an elevated suction reservoir (Reservoir A) and forcing it to a higher discharge destination.



Let us map out the elevations relative to the designated pump centerline datum ($z_{pump} = 0$):

- **Suction Reservoir Level (z1):** +6m (above centerline)
- **Discharge Reservoir Level (z2):** +55m (above centerline)

The pump moves a volumetric flow rate (Q) of 0.3 m^3/s through a suction line with a diameter (D_1) of 110 mm (0.110 m) and a discharge line with a diameter (D_2) of 95 mm (0.095 m). The total friction and minor head losses (h_L) in the piping network are limited to 45% (0.45) of the velocity head inside the suction line.

By applying the steady-flow energy equation between the suction boundary (1) and discharge boundary (2), the absolute TDH is broken down into its static elevation head (h_s), kinetic velocity head (h_v), static pressure head (h_p), and total head loss (h_L):

$$TDH = hs + h_v + h_p + h_L$$

Because both reservoirs are large bodies of liquid open directly to the ambient atmosphere, the static surface pressures are equal ($P_1 = P_2 = P_{atm}$), making the net pressure head component zero ($h_p = 0$).

Step 1: Calculate Fluid Flow Velocities

Using the **Continuity Equation** (Q = Av), compute the independent fluid stream velocities inside the suction and discharge pipes:

- **Suction Velocity (v_1):**

$$Q = A_1 v_1$$

$$0.3\ \frac{m^3}{s} = \left[\frac{\pi}{4}(0.110\ m)^2\right] v_1$$

$$v_1 = 31.5679\ \frac{m}{s}$$

- **Discharge Velocity (v_2):**

$$Q = A_2 v_2$$

$$0.3\ \frac{m^3}{s} = \left[\frac{\pi}{4}(0.095\ m)^2\right] v_2$$

$$v_2 = 42.3238\ \frac{m}{s}$$

Note: Standard industrial water pipes restrict velocity to a maximum of 2 to 6 m/s} to prevent destructive water hammer and massive friction drops. These exceptionally high velocities indicate that this problem was written purely as an academic calculation exercise.

<u>Step 2:</u> Evaluate Individual Head Components

Now evaluate each independent energy contribution to determine the total head demands:

- **Static Elevation Head (h_s):**

$$h_s = z_2 - z_1 = (55 - 6)m = 49\ m$$

- **Net Kinetic Velocity Head (h_v):**

$$h_v = \frac{v_2^2 - v_1^2}{2g_o} = \frac{(42.3238\ m/s)^2 - (31.5679\ m/s)^2}{2\left(9.81\ \frac{m}{s^2}\right)} = 40.5082\ m$$

- **Friction and Minor System Head Losses (h_L):**

$$h_L = 0.45\left(\frac{v_1^2}{2g_o}\right) = 0.45\left[\frac{(31.5679\ m/s)^2}{2\left(9.81\ \frac{m}{s^2}\right)}\right] = 22.8562\ m$$

<u>Step 3:</u> Compute Total Dynamic Head (TDH)

Sum up the calculated components to find the total head the pump must generate:

TDH = 49m + 40.5082m + 22.8562m = 112.3644m

<u>Step 4:</u> Determine Fluid Water Power (WP) and Shaft Brake Power (BP)

Using a standard water specific weight (γ_W = 9.81 kN/m^3), calculate the net power transferred to the fluid (WP):

$$WP = Q\gamma_W TDH = \left(0.3\frac{m^3}{s}\right)\left(9.81\frac{kN}{m^3}\right)(112.3644m) = 330.6884\, kW$$

To find the actual mechanical brake power (BP) supplied by the motor shaft, divide by the pump's hydraulic efficiency ($\eta_V = 80\%$):

$$BP = \frac{WP}{\eta_V} = \frac{330.6884\, kW}{0.80} = 413.3605\, kW$$

<u>Step 5:</u> Extract Mechanical Shaft Torque (T)

Brake power is the product of rotational angular velocity and shaft torque (BP = $\omega \cdot T$). For a system spinning at n = 380 rpm, torque is calculated using the standard rotational power relationship:

$$BP = \frac{2\pi \cdot T \cdot N}{60}$$

$$413.3605\, kW = \frac{2\pi \cdot T \cdot (380\, rpm)}{60}$$

T = 10.3879 kN-m

The calculated pipeline velocities ($v_1 = 31.5679\,\frac{m}{s}$ and $v_2 = 42.3238\,\frac{m}{s}$) are extremely high for liquid water systems. In a real industrial plant, fluid moving at these speeds would experience extreme friction losses.

More importantly, according to Bernoulli's principle, such high velocities create a massive pressure drop at the suction inlet. This drop would drop the local pressure below the vapor pressure of water, causing instant, severe cavitation that would quickly destroy the pump impeller. This problem serves as an excellent reminder of why real-world design standards limit liquid pipe velocities to much lower, safer ranges.