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ABE 304

Pump Configurations and Flow through a Pipe

Background on centrifugal pump operation and operating principle

Centrifugal pumps are the most common types of pumps found in agriculture, water processing facilities, power plants, petroleum industries, and many other major industrial processes. The primary operating principle of a centrifugal pump is centrifugal force. Inside of the pump, fluid is drawn into the inlet, or suction port, that is powered by a shaft-driven impeller. At the suction port of the pump, the pressure is relatively low compared to the rest of the system. The low pressure allows liquid to be pulled in from the reservoir. As the pump rotates, the liquid is moved radially outwards along the vanes of the impeller. As it is cast outwards, it collects in the outer casing with a volute flow pattern, and the velocity of the fluid significantly increases. The liquid is then discharged from the pump, and it has a higher pressure than when it first entered due to the increase in velocity caused by centrifugal force. The most advantageous feature of centrifugal pumps is that their flow rates can be inexpensively and simply throttled over large ranges of flow rates without damaging the pump. This is due to the fact that the flow rate of the pump will naturally vary with total dynamic head (TDH) changes in the piping system. This principle can be exploited by using a valve at the discharge pipe to make changes in the TDH of the system and in the flow rate of the centrifugal pump. Another noteworthy operational feature is that centrifugal pumps can run at lower RPMs with the same throughput compared to their counterparts, thus there is less wear on the pump. While this extends the life of the pump, it does make the machinery cost more, as larger casings and impellers are needed to compensate for the rotational speeds.

Importance of calculating friction losses in a pumping-piping system

Friction is a significant factor in calculating fluid flow properties in pipe systems. While it is sometimes neglected to simplify modeling, it is useful when calculating realistic models. Normally, friction is accounted for in elbows, valves, fittings, and total pipe length by using well-established/averaged literature values for these components. Calculating frictional losses can help with modeling discharge pressure requirements for pumps, energy requirements for pumping-piping systems, pump efficiency, and theoretical maximum throughputs of a pumping-piping system. Without considering frictional losses, all energy models of these systems would be less accurate.

Equations necessary for analysis

In using a pump in a flow system, the underlying principle is the conservation of mechanical energy, meaning that the mechanical energy created by the pump acts on the fluid and is converted into flow, kinetic, and potential energy. Equation 1 expresses this principle.

$$e_{mech} = \frac{P}{\rho} + \frac{v^2}{2} + gz \tag{1}$$

P represents the pressure, ρ represents the density of the fluid, *v* represents the velocity of the fluid, *g* represents the gravitational constant, and *z* represents the height of the fluid. *P*, *v*, and *z* can be expressed as the differences in these quantities between two points in the system. In reality, however, all of the work done by the pump is not converted into the fluid's mechanical energy. The efficiency of the pump is expressed in Equation 2.

$$\eta_{pump} = \frac{e_{mech,fluid}}{W_{shaft,in}} \tag{2}$$

Work done by the pump is represented as $W_{shaft,\ in}$.

For an adiabatic system with a steady, incompressible fluid, the Bernoulli equation (Equation 3) approximately relates the pressure, velocity, and elevation of a fluid.

$$constant = \frac{P}{\rho} + \frac{v^2}{2} + gz \tag{3}$$

By dividing the Bernoulli equation by the gravitational constant, each term is expressed in length units and called "heads". The pressure or head increase created by the work done by the pump can be calculated with Equation 4.

$$\Delta h = a - bQ^2 \tag{4}$$

In this equation, *a* and *b* are constants specific to the pump and *Q* is the volumetric flow rate. This equation changes for different configurations of multiple pumps. Pumps in series create additive pressure increases (Equation 5) and parallel pumps divide the flow rates (Equation 6), assuming that the pump-specific constants are equal in each pump.

$$\Delta h = n(a - bQ^2) \tag{5}$$

$$\Delta h = a - b(\frac{Q}{n})^2 \tag{6}$$

In the above equations, *n* represents the number of pumps in the respective configurations. Equations 4, 5, and 6 can be expressed in a pump performance curve by graphing the head difference versus the flow rate.

The biggest source of head loss in a system is friction, as was explained in the previous section. The Fanning friction factor is the quotient of shear stress at the wall (τ_s) and the product of the fluid's density and velocity head. There are different friction factor equations for laminar and turbulent flow, shown in Equations 7 and 8, respectively.

$$f = \frac{16}{Re} \tag{7}$$

$$f = \frac{2\tau_s}{\rho v^2} \tag{8}$$

Re in Equation 7 represents the Reynolds Number (Equation 9), a dimensionless parameter which determines whether flow is turbulent or laminar.

$$Re = \frac{\rho v L}{\mu} \tag{9}$$

L represents the inner diameter of the pipe and μ is the viscosity of the fluid. The Reynolds Number can be related to the velocity and friction in a smooth pipe. This relationship is presented visually in the form of a Moody Diagram, which graphs the friction factor and relative roughness on opposite sides of the y-axis versus the Reynolds Number, by solving for the friction factor for a set of Reynolds Numbers and relative roughness of pipe constants (ε/D). The Colebrook Equation (Equation 10) is used to create the Moody Diagram.

$$\frac{1}{\sqrt{f}} = -2log_{10}\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{Re\sqrt{f}}\right) \tag{10}$$