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Velocity Head

Two of the most useful and basic equations are

$$\Delta h = \frac{u^2}{2g} \quad (1)$$

$$\Delta P(V) + \frac{\Delta u^2}{2g} + \Delta Z + E = 0 \quad (2)$$

where

Δh = Head loss in feet of flowing fluid

u = Velocity in ft/sec

$g = 32.2 \text{ ft/sec}^2$

P = Pressure in lb/ft²

V = Specific volume in ft³/lb

Z = Elevation in feet

E = Head loss due to friction in feet of flowing fluid

Applications

In Equation 1 Δh is called the “velocity head.” This expression has a wide range of utility not appreciated by many. It is used “as is” for

1. Sizing the holes in a sparger
2. Calculating leakage through a small hole
3. Sizing a restriction orifice
4. Calculating the flow with a pilot tube

With a coefficient it is used for

1. Orifice calculations
2. Relating fitting losses, etc.

Why a Coefficient?

For a sparger consisting of a large pipe having small holes drilled along its length Equation 1 applies directly. This is because the hole diameter and the length of fluid travel passing through the hole are similar dimensions. An orifice, on the other hand, needs a coefficient in Equation 1 because hole diameter is a much larger dimension than length of travel (say 1/8 in for many orifices). Orifices will be discussed under “Metering” in this chapter.

Sonic Velocity

For the situations covered here, compressible fluids might reach sonic velocity. When this happens, further decreases in downstream pressure do not produce additional flow. Sonic velocity occurs at an upstream to downstream absolute pressure ratio of about 2 : 1. This is shown by the formula for sonic velocity across a nozzle or orifice.

critical pressure ratio = $P_2/P_1 = [2/(K+1)]^{K/(K-1)}$
when $K = 1.4$, ratio = 0.528, so $P_1/P_2 = 1.89$

To determine sonic velocity, use

$$V_s = (KgRT)^{0.5}$$

where

V_s = Sonic velocity, ft/sec

$K = C_p/C_v$, the ratio of specific heats at constant pressure to constant volume

$g = 32.2 \text{ ft/sec}^2$

$R = 1,544/\text{mol.wt.}$

T = Absolute temperature, °R

P_1, P_2 = Inlet, outlet pressures, psia

Critical flow due to sonic velocity has practically no application to liquids. The speed of sound in liquids is very high.

For sonic velocity in piping see the section on “Compressible Flow.”

Bernoulli Equation

Still more mileage can be gotten out of $\Delta h = u^2/2g$ when using it with Equation 2, which is the famous Bernoulli equation. The terms are

1. The PV change
2. The kinetic energy change or “velocity head”
3. The elevation change
4. The friction loss

These contribute to the flowing head loss in a pipe. However, there are many situations where by chance, or

Here is a quick check for water settling.

1. Estimate the water terminal settling velocity using:

$$U_T = 44.7 \times 10^{-8} (\rho_w - \rho_o) F_s / \mu_o$$

where

U_T = Terminal settling velocity, ft/sec

F_s = Correction factor for hindered settling

ρ_w, ρ_o = Density of water or oil, lb/ft³

μ_o = Absolute viscosity of oil, lb/ft-sec

This assumes a droplet diameter of 0.0005 ft. F_s is determined from:

$$F_s = X^2 / 10^{1.82(1-X)}$$

where

X = Vol. fraction of oil

2. Calculate the modified Reynolds number, Re from:

$$Re = 5 \times 10^{-4} \rho_o U_T / \mu_o \quad (\text{usually} < 1.0)$$

This assumes a droplet diameter of 0.0005 ft

3. Calculate U_s/U_T from:

$$U_s/U_T = A + B(\ln Re) + C(\ln Re)^2 + D(\ln Re)^3 + E(\ln Re)^4$$

where

U_s = Actual settling velocity, ft/sec

$A = 0.919832$

$B = -0.091353$

$C = -0.017157$

$D = 0.0029258$

$E = -0.00011591$

4. Calculate the length of the settling section as:

$$L = hQ / AU_s$$

where

L = Length of settling zone, ft

h = Height of oil, ft

Q = Flow rate, ft³/sec

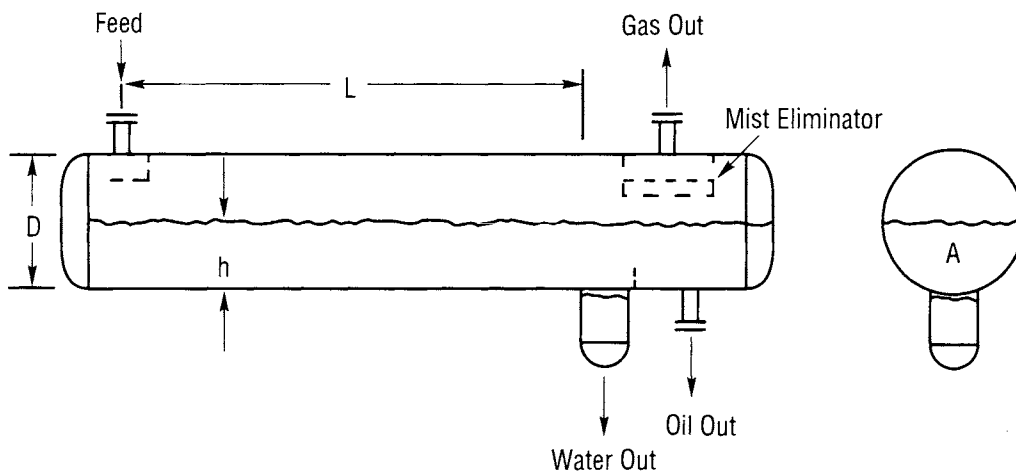
A = Cross-sectional area of the oil settling zone, ft²

This allows the water to fall out and be drawn off at the bootleg before leaving the settling section.

Vessel Example

Given:

Rates	<u>mols/hr</u>
Feed	5200
Liquid	5000
Vapor	200



Process Efficiency

Because fuel costs are high, the search is on for processes with higher thermal efficiency and for ways to improve efficiencies of existing processes. One process being emphasized for its high efficiency is the gas turbine "combined cycle." The gas turbine exhaust heat makes steam in a waste heat boiler. The steam drives turbines, often used as helper turbines. References 1, 2, and 3 treat this subject and mention alternate equipment hookups, some in conjunction with coal gasification plants.

Arrangements that combine the gas turbine, steam helper turbine, and electric generator on a single shaft are gaining acceptance.⁴ Already established overseas, these are gaining a foothold in the U.S. Advantages are the possibility of lower first cost, operating simplicity, a common lubricating oil system for the steam turbines and electric generator, and a smaller site footprint. In addition, the heat recovery steam generator bypass stack is eliminated. On multishaft units the bypass stack allows phased construction and simple-cycle operation. The single shaft units can employ a synchronous clutch between the generator and steam turbine(s). This permits startup in simple-cycle mode while warming the steam systems. The clutch engages when the steam turbine shaft rotating speed reaches that of the generator. Many plants that must shut-down and restart daily are using the synchronous clutch.

Reference 5 is a well-written report which discusses power plant coal utilization in great detail. It gives a thermal efficiency of 80–83% for modern steam generation plants and 37–38% thermal efficiency for modern power generating plants at base load (about 70%). A modern base load plant designed for about 400 MW and up will run at steam pressures of 2,400 or 3,600 psi and 1,000°F with reheat to 1,000°F and regenerative heating of feedwater by steam extracted from the turbine. A thermal efficiency of 40% can be had from such a plant at full load, and of 38% at high annual load factor. The 3,600 psi case is supercritical and is called a once-through-boiler since it has no steam drum. Plants designed for about 100–350 MW run around 1,800 psi and 1,000°F with reheat to 1,000°F. Below 100 MW, a typical condition would be about 1,350 psi and 950°F with no reheat. Reference 5 states that below 60% load factor, efficiency falls off rapidly and that the average efficiency for all steam power plants on an annual basis is about 33%.

For any process converting heat energy to mechanical efficiency, the Carnot efficiency is the theoretical maximum. It is calculated as

$$\frac{T_1 - T_2}{T_1} \times 100 \quad (1)$$

where

T_1 = Temperature of the heat source, °R

T_2 = Temperature of the receiver where heat is rejected, °R

Therefore, the efficiency is raised by increasing the source temperature and decreasing the receiver temperature.

The efficiency for a boiler or heater is improved by lowering its stack temperature. The stack minimum temperature is frequently limited by SO₃ gas dew point. References 6, 7, and 8 discuss this important subject. A stack as hot as 400°F (or perhaps higher) can have problems if the SO₃ concentration is high enough. Reference 9 states that SO₃ condensation will produce a blue-gray haze when viewed against a clear blue sky.

A very useful relationship for determining the maximum available energy in a working fluid is

$$\Delta B = \Delta H - T_o \Delta S \quad (2)$$

where

ΔB = Maximum available energy in Btu/lb

ΔH = Enthalpy difference between the source and receiver, Btu/lb. For a typical condensing steam turbine it would be the difference between the inlet steam and the liquid condensate

T_o = Receiver temperature, °R

ΔS = Entropy difference between the source and receiver, Btu/lb °F

To obtain lb/hr-hp, make the following division:

$$\frac{2,545}{\Delta B}$$

Equation 2 will yield the same result as the Theoretical Steam Rate Tables (Reference 10). Therefore, this is a handy way of getting theoretical steam rates when only a set of steam tables sans Mollier diagram are available.

Source

1. Moore, R., and Branan, C., "Status of Burnham Coal Gasification Project," *Proceedings, 54th Annual Convention, Gas Processors Association*, Houston, Texas, March 10–12, 1975.

Summary and Conclusions

Shortcut equipment design methods are part of an engineering designer's life. In addition to using such methods frequently and in all of his designs, he should always be on the alert for chances to develop his own for future generations to use. The opportunities for doing this are great and the results can be very rewarding.

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