FLUID STATICS AND PRESSURE MEASUREMENT

This chapter deals with the principles of *hydrostatics* and methods of measuring pressure. Hydrostatics deals with the properties and characteristics of liquids that are not in motion. When dealing with stationary gases the subject is called *pneumatics*. In keeping with the general theme of the book, only incompressible fluids are discussed here. Basic definitions applicable to problem solving and flow measurement are presented.

PRESSURE DEFINITIONS

If a force F(N) is uniformly distributed over a flat surface of area A(m^2), then the pressure is P=F/A(N/m^2 or lb_f/ft^2). As shown, the units of pressure are force per unit area. A *normal pressure* acts only in a direction perpendicular to the surface sustaining it. The term pressure can take on several meanings.

For fluids, *relative and absolute* pressures are defined in terms of a chosen reference. Relative pressure is the measured pressure with respect to the atmosphere. Absolute pressure is that measured with respect to a complete vacuum. In the former, relative pressure is dependent on location, temperature, humidity and other environmental factors. Since absolute pressure is referenced with respect to a complete vacuum, it is a fixed measurement or true measurement.

Barometric pressure refers to a measurement of the atmospheric pressure using a barometer.

Gauge pressure is that pressure expressed as a quantity measured from above some reference pressure, which is usually atmospheric. The relationship of gauge, barometric and absolute pressure is as follows:

$$\text{Gauge pressure} + \text{Barometric Pressure} = \text{Absolute Pressure} \tag{1}$$

Figure 2 defines the pressure relationships graphically along with common engineering units.

The terms *hydrostatic* or *static pressure* at a point in a fluid refer to the compressive stress at that point. Consider a fluid at rest in a containing vessel. Hydrostatic pressure is the stress normal to any surface exerted by the fluid. At any given point it has the same magnitude, regardless of the orientation of the surface. Hence, the pressure exerted on the containing walls of the vessel by a stationary fluid is normal and equal to, at each point, the static pressure of the adjacent fluid. In the case of a flowing fluid, the pressure on the containing surface is not wholly normal.

Consider now a differential cube of fluid. The variation of the static pressure in any direction in the fluid may be found from Newton's law:

$$F = (\text{mass} \times \text{acceleration})/g_c \tag{2}$$

If there is no relative motion, the only forces acting on the boundaries of the cube are gravity and the normal outward pressure. For forces acting in the x and y directions.

$$\rho x - \frac{\partial P}{\partial x} = \frac{\rho a_x}{g_c} \tag{3a}$$

$$\rho y - \frac{\partial P}{\partial y} = \frac{\rho a_y}{g_c} \tag{3b}$$

Where ρ = fluid density, lb_m/ft^3

X, Y = x and y component of extraneous forces (lb_f/lb_m), respectively

a_x, a_y = x and y component of the fluids acceleration (lb_f/lb_m), respectively

$g_c = 32.1740\ (lb_m)(ft)(lb_f)(s^2)$.

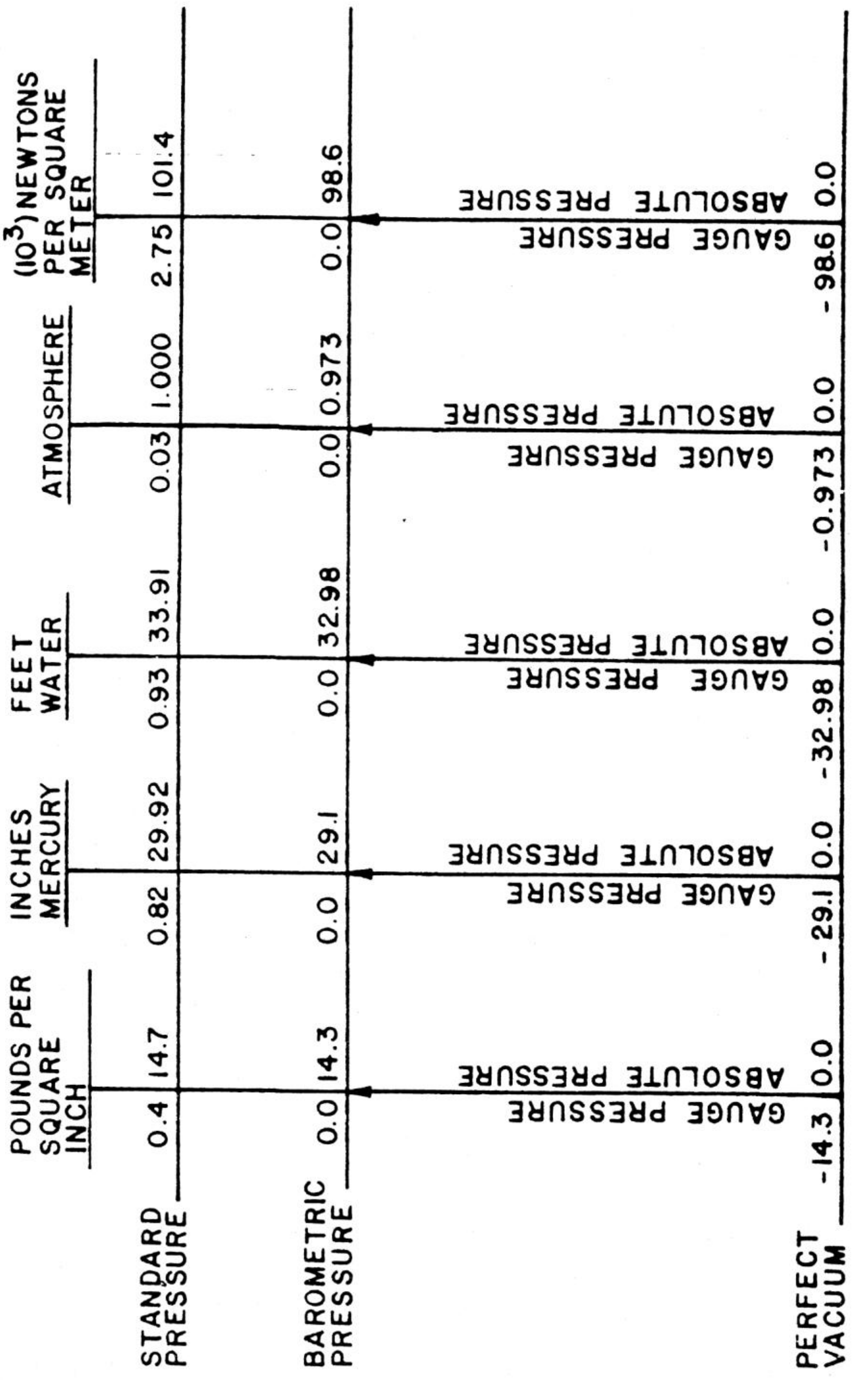

Figure 2. Shows the relationship of standard, barometric and vacuum pressure for different engineering units. Standard atmosphere is the pressure obtained in a standard gravitational field and is equivalent to 14.696 psi or 760 mm mercury at 0°C. Atmospheric pressure is a variable that must be measured with a barometer each time.

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Since y is measured vertically downward, $Y=g/g_c$; where g is the local acceleration due to gravity (ft/s). For the horizontal x direction, the force component X is zero.

Pressure will vary with depth, y, for a stationary fluid accordingly:

$$\frac{dP}{dy} = \frac{\rho g}{g_c} \tag{4}$$

If the density of the fluid is constant, then the pressure is the same everywhere at the same depth. These conclusions lead to Pascal's law, which for incompressible fluids states that any increase in the pressure at any point is accompanied by an equal increase at every point in the fluid.

If our differential fluid cubical is now extended to a column, then it can be said that the static pressure at the base of the column is greater than the top surface by an amount $H\rho\ g/g_c$(N/m^2 or lb_f/ft^2) where H is the *head* of the column. The term *head* refers to the height of the fluid above some datum line. The magnitude of the pressure differential can be specified by measuring the head of the column of fluid that results in an increase in the static pressure equal in magnitude to the pressure described. Fluid heads generally measure gauge pressure.

GENERAL PROPERTIES OF LIQUIDS

Liquids are simply the *condensed phases* of gases (so are solids for that matter). The term "condensed phases" emphasizes the high density of the liquid as compared with the low density of gases. Consider a vessel designed to contain an amount of oxygen not to exceed a few hundred pounds mass: the weight of water required to fill the same vessel would be several hundred tons. Another way of saying this is that the volume per mole of gases is very large, whereas for liquids it is quite small. At conditions of standard temperature and pressure (STP) any gas occupies 22,400 cm^3/mol. Most liquids occupy between 10 and 100 cm^3/mol. Hence, the molar volume of a liquid is 500-1000 times smaller than that of a gas.

Since the volume ratio of gas to liquid is as high as 1000, the ratio of distances between molecules in the gas as compared to the liquid is $\sqrt[3]{1000}$, or 10. That is, the distance between gas molecules is ten times farther apart than those of a liquid. In liquids, the distance between molecules is on the order of the molecular diameter. This large difference in molecular distances between gases and liquids accounts for the drastic difference in properties between the two states of matter. Intermolecular forces (i.e. van der Waals forces) have a relatively short field of influence such that the effect of these forces decreases dramatically with increasing distances between molecules. In fact, the magnitude of van der Waals forces in liquids is on the order of 10^6 times larger than it is in gases because of the shorter distances between molecules.

Because molecular distances are so small and intermolecular forces correspondingly so large for liquids, the properties of liquids depend greatly on the details of the forces acting between molecules. Hence, there are no simple relations or a universal equation of state to describe liquid behaviour as there are for gases. Hence, properties of liquids vary greatly whereas generalities can be made about many of the physical properties of gases.

One example is the dependence of volume on temperature. For liquids, volume varies with temperature at constant pressure by an expression of the following form:

$$V = V_0\left(1+kT\right) \tag{34}$$

V_0 is the volume of the liquid at some reference temperature (usually 0℃) and k is the coefficient of the thermal expansion. Gases follow a general dependency similar to Equation 34; however, k is approximately constant for all gases. For liquids, k depends on the specific liquid.

V_0 in Equation 34 is a function of pressure and follows the relationship

$$V_0 = \hat{V}_0\left[1-\beta\left(p-1\right)\right] \tag{35}$$

Where $\hat{V}_0$ is the volume at STP and β is the coefficient of compressibility. Parameter β is observed to be constant over a wide range of pressures for a particular material but is different for each substance and for the solid and liquid states of the same material.

Equation 35 states that the volume of a liquid decreases linearly with pressure. For gases, volume is inversely proportional to pressure. Even more dramatic a contrast is that β for liquids is typically 10^{-6} atm, whereas for gases it is significant. As an example, if we subject water to a pressure change from 1 to 2 atm, Equation 35 predicts less than 10^{-3}% reduction in volume. If a sample of air were subjected to the same pressure change, its volume would be reduced by more than a factor of two. Because of this characteristic of insignificant volume changes from moderate pressure changes, liquids are referred to as *incompressible fluids*. That is, for practical purposes, values of β are taken to be zero for liquids.

If we go back to our vessel discussed earlier, evacuate the gas and place a known quantity of a pure liquid in it, then a portion of the liquid will evaporate so as to fill the remaining volume of the vessel with vapour. After equilibrium is established, the pressure exerted by the vapour in the vessel becomes dependent on the temperature of the system only. This pressure developed is the vapour pressure of the liquid and will increase rapidly with temperature. The normal boiling point of the liquid is the temperature at which the vapour pressure becomes equal to the atmospheric pressure.

The amount of heat absorbed during the transformation of liquid to vapour state is the *heat of vaporization*. For vaporization to occur, liquid molecules are pulled apart against intermolecular forces. The energy needed to accomplish this is measured by the heat of vaporization. From thermodynamics, the relationship between vapour pressure and the heat of vaporization is

$$\ln P_V = -\frac{\dot{Q}_{vap}}{RT} + \ln P_\infty \tag{36}$$

Where $\dot{Q}_{vap}$ is the heat of vaporization, R is the gas law constant, T is absolute temperature, and P_∞ is a constant defined as

$$P_\infty = \left(atmos.press.\right)\exp\left(+\dot{Q}_{vap} / RT_b\right) \tag{37}$$

Where T_b is the boiling point of the liquid.

Equation 36 is one form of the Clausius-Clapeyron equation. As an aside, a plot of log P_V versus 1/T produces a straight line whose slope is $-\dot{Q}_{vap} / R$. By obtaining measurements of the vapour pressure at various temperatures and preparing such a plot, the heat of vaporization can be readily obtained for the liquid.

There are two other general properties of interest to our forthcoming discussions. One is interfacial tension, already described in the previous chapter, and the other is the coefficient of viscosity. As we will see, viscosity is simply a property of the fluid, which results in the resistance to flow when the fluid is subjected to stress. Because the molecules of a liquid are packed more closely, a liquid is more viscous than a gas. Both the close molecular spacing and high intermolecular forces contribute to this resistance of flow.

MOLECULAR THEORIES OF VISCOSITY

Let us consider a fluid (either gas or liquid) sandwiched between two flat plates located at a distance, y, apart. If the bottom plate is fixed and a force, F, is applied to the upper plate to set it in motion at a constant velocity, u, in the x-direction, the following sequence of events occurs:

1. At time t=0, the fluid has a velocity profile that is flat and vertical (see Figure 9(A)).
2. At a small increment of time to t_1, the velocity of the fluid goes through a transient or unsteady flow where the momentum of the fluid builds up (refer to Figure 9(B)).
3. Finally, at some large time, t_2, steady flow is achieved and the velocity distribution appears as in Figure 9(C).

The force per unit area applied to the upper plate (assuming the flow is laminar) is proportional to the velocity increase in distance, y (taking y as positive in the upward direction). This is defined by the following relation, where the constant of proportionality, μ , is the viscosity of the fluid:

$$\frac{F}{A} = \mu \frac{du}{dy} \tag{38}$$

The value of μ describes a physical property of the fluid that establishes the magnitude of resistance to flow. We shall first discuss the property of viscosity on a molecular scale and then relate this to macroscopic observations. Since properties of gases and liquids vary greatly, their molecular behaviour is quite different and therefore should be discussed separately.

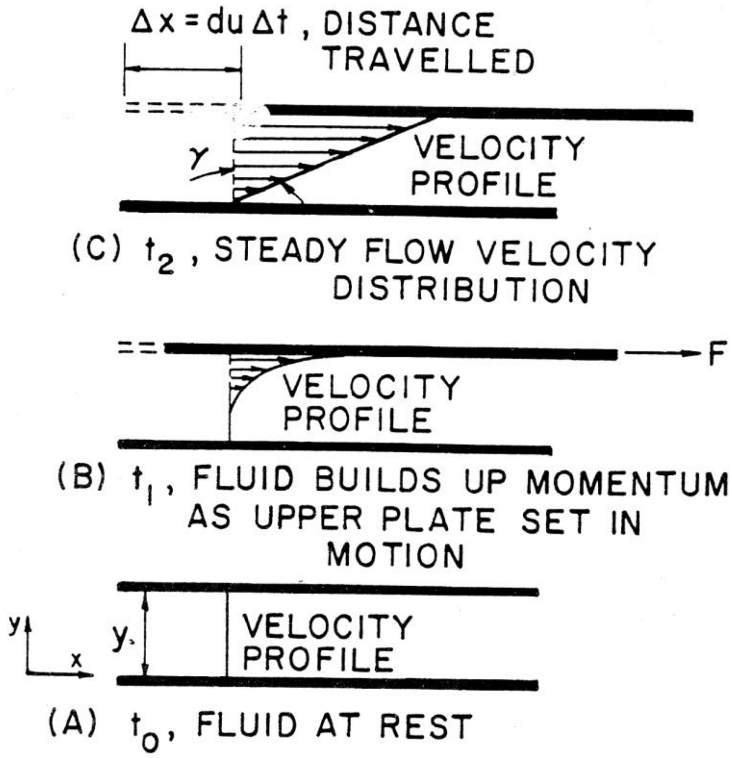


Figure 9. Illustrates the development of steady laminar flow.

PROPERTIES OF NEWTONIAN FLUIDS

Equation 38 is our governing expression for describing the laminar flow situation shown in Figure 9 and, from a macroscopic viewpoint, is Newton's law of viscosity, where $\tau_{yx} = F / A$.

$$\tau_{yx} = F / A = \mu \frac{du}{dy} \tag{52a}$$

Or, if English units are used,

$$\tau_{yx} = \frac{\mu}{g_c} \frac{du}{dy} \tag{52b}$$

The conversion factor, g_c [value of 32.174 (lb_m/lb_f) (ft/s^2)] is introduced because of the dual system of units we shall adhere to in our computations. Careful checks for dimensional consistency are needed in all practical calculations when dealing in English units. If SI units are employed the value of g_c is unity.

In writing Equation 52 we have assumed the ideal case of zero slip condition at the plate wall. The x-distance traveled by a layer of fluid immediately beneath the top plate during time Δt will be $du\Delta t$. Note the angle, γ, shown in Figure 9(C) provides an indication of the deformation of the fluid caused by the imposed stress, τ_{yx}. For small values of γ,

$$\gamma = \frac{du\Delta t}{dy} \tag{53}$$

The treatment of a fluid undergoing deformation is analogous to the deformations in elastic solids, where γ is referred to as the strain. In our case, the rate of deformation is the same as the velocity gradient:

$$\frac{du}{dy} = \frac{d\gamma}{dt} \tag{54}$$

The more exact name for *du/dy* used by rheologists is the *shear rate* of the fluid.

Many textbooks give an alternate interpretation of the system shown in Figure 9, where y is taken to be positive in the downward direction. Then Equation 52 and 54 become

$$\tau_{yx} = -\frac{\mu}{g_c}\frac{du}{dy} \tag{55}$$

And

$$\frac{d\gamma}{dt} = -\frac{du}{dy} \tag{56}$$

For an incompressible fluid, Equation 55 is more properly stated as

$$\tau_{yx} = -\frac{\mu}{\rho g_c}\frac{d(u\rho)}{dy} \tag{57}$$

The quantity of $u\rho$ is the *momentum concentration* (has units of momentum per unit volume). Hence, $d(u\rho)/dy$ is the momentum concentration gradient in the y-direction and is negative according to the sign convention for the y-direction. The quantity $\tau_{yx} g_c$ may be considered to be the flux of the x-directed momentum in the y-direction.

A Newtonian material is simply any substance whose shear rate is directly proportional to the stress or shearing force applied to the fluid. Equation 57 is a straight line, with the intercept at the origin and whose slope is μ . (A plot of τ_{yx} versus *du/dy* is referred to as flow curve.) Hence, viscosity is a constant, independent of the shear rate, and only depends on the process temperature and pressure of the particular fluid. The energy dissipated by the deformation of these type fluids results from the collisions of relatively small molecules. Hence, virtually all gases, as well as liquids of low-molecular-weight, display Newtonian behavior.

THE BERNOULLI EQUATION

The principal equation for computing pressure drop for flowing liquids in pipes and fittings is the generalized Bernoulli equation:

$$\underbrace{-\frac{\Delta P}{\rho}}_{\substack{\text{pressure}\\\text{change}}} = \underbrace{\frac{\alpha\Delta(u^2)}{2g_c}}_{\substack{\text{kinetic}\\\text{energy}\\\text{change}}} + \underbrace{\frac{g\Delta Z}{g_c}}_{\substack{\text{elevation}\\\text{change}}} + \underbrace{F}_{\substack{\text{frictional}\\\text{head loss}}} \qquad (1)$$

		(English Units)	(cgs Units)
where	F = frictional head loss	$ft\text{-}lb_f/lb_m$	$Pa \cdot m^3/kg$
	g = acceleration of gravity	ft/s^2	m/s^2
	g_c = conversion factor	$32.174\ ft \cdot lb_m/lb_f\text{-}s^2$	
	ΔP = pressure change	lb_f/ft^2	Pa
	u = fluid velocity	ft/s	m/s
	Z = elevation	ft	m
	ρ = fluid density	lb_m/ft^3	kg/m^3

α = kinetic energy correction term whose value depends on the velocity profile (for turbulent flow $\alpha = 1.1$; for laminar flow $\alpha = 2.0$)

Most design equations for single-phase flow can be derived from the generalized Bernoulli equation. The importance of each term in the equation varies with application.

For Newtonian liquid flow, constant properties (viscosity and density) can be assumed. Exceptions to this rule are non-Newtonian liquids and non-isothermal flow. In non-isothermal flow, properties change because of either heat exchange, or heat production or consumption in the fluid by chemical reaction or friction losses.

If isothermal conditions exist across the pipe cross section but not along the flow axis, pressure drop can be computed by dividing the pipe into a number of small lengths and calculating the pressure drop in each section.

THE CONTINUITY EQUATION

The continuity equation mathematically expresses the law of conservation of mass. Figure 2 represents a differential element of fluid flowing in a three dimensional space. The fluid element has dimensions dx, dy, dz. Within this orientation, the mass flow of fluid through the cubical element can be derived. Fluid

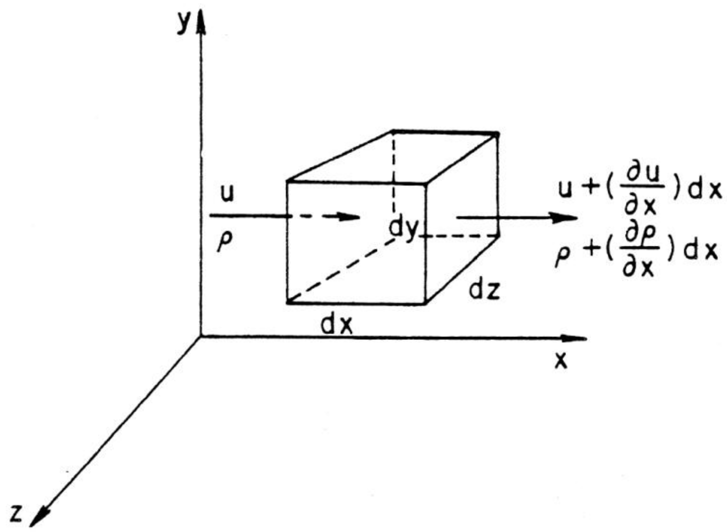


Figure 2. One-dimensional flow of a cubical fluid element.

velocity, u, and density, ρ, represent conditions on the x face of the fluid cube. At the x+dx position, the velocity and density are

$$u + (\partial u / \partial x) dx \tag{9}$$

and

$$\rho + (\partial \rho / \partial x) dx \tag{10}$$

respectively, $\partial u / \partial x$ and $\partial \rho / \partial x$ represent the rate of change of velocity and density with respect to the x-axis. The variables *u* and ρ are dependent, while x as well as time are independent variables.

For a differential time, dt, a mass balance can be developed:

$$\text{Mass – Input to x-face} = \rho\ udydzdt \tag{11}$$

$$\text{Mass – Output from x+dx-face} = \left(\rho + \frac{\partial \rho}{\partial x}dx\right)\left(u + \frac{\partial u}{\partial x}dx\right)dydzdt \tag{12}$$

By neglecting the dx^2 term in Equation 12 we obtain:

$$\left(\rho u + u\frac{\partial \rho}{\partial x}dx + \rho\frac{\partial u}{\partial x}d\right)dydzdt = \text{Mass – Output} \tag{13}$$

A simple mass balance expression states

$$\text{Input} = \text{Output} + \text{Accumulation} \tag{14}$$

Where

$$\rho udydzdt = \left(\rho u + u\frac{\partial \rho}{\partial x}dx + \rho\frac{\partial u}{\partial x}dx\right)dydzdt + \frac{\partial \rho}{\partial t}dxdydzdt \tag{15}$$

Where $\left(\partial \rho / \partial t\right)dxdydzdt$ represents the accumulation of mass in the fluid element.

The general material balance can be simplified to

$$-u\frac{\partial \rho}{\partial x} - \rho\frac{\partial u}{\partial x} = \frac{\partial \rho}{\partial t} \tag{16}$$

The left-hand side (LHS) of this equation is an exact derivative. The continuity equation for one-dimensional flow is stated as follows:

$$-\frac{\partial\left(\rho u\right)}{\partial x} = \frac{\partial \rho}{\partial t} \tag{17}$$

This is the law of conservation of mass of a fluid flowing parallel to the x-axis. That is, Equation 17 describes the case of one-dimensional flow in which both the velocity and fluid density are functions of only x and t.

A similar analysis can be used to develop the equation of continuity for flow in three dimensions. Knudsen and Katz [4] and Bird et al. [5] perform this analysis in detail. Table 4 summarizes the various forms of the continuity equation for both compressible and incompressible fluids under steady and unsteady state flow conditions.

MOMENTUM EQUATIONS

Newton's second law of motion states that the time rate of change of momentum is proportional to external forces. That is,

$$\text{Mass} \times \text{Acceleration} = \text{External Force} \tag{18}$$

or

$$m\frac{du}{dt} = Fg_c \tag{19}$$

The product on the LHS of Equation 19 is referred to as the inertial force. In fluid flow systems inertial forces act in three directions. The resultant inertial force can be related to the external forces acting on the fluid in motion. There are three major external forces that can produce fluid motion, namely, gravity forces, normal or pressure forces, and shear forces. Shear forces result from the resistance of the fluid to undergo deformation.

For the general case of three-dimensional flow where there are three component velocities, u, v, and w, an expression for unsteady flow for the x-direction is

$$du = \frac{\partial u}{\partial x}dx + \frac{\partial u}{\partial y}dy + \frac{\partial u}{\partial z}dz + \frac{\partial u}{\partial t}dt \tag{20}$$

Dividing this expression by dt and noting that u=dx/dt, v=dy/dt and w=dz/dt,

$$\frac{du}{dt} = u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} + \frac{\partial u}{\partial t} \tag{21}$$

The term du/dt represents the fluid's acceleration in the x-coordinate direction.

The inertial force in the x-direction is the product of mass and acceleration; then, defining mass as

$$m=\rho \ dxdydz \tag{22}$$

Using Equation 21, the inertial force in the x-direction becomes

$$F_x = \frac{\rho}{g_c}\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} + \frac{\partial u}{\partial t} \right) dxdydz \tag{23}$$

To apply Newton's second law, the inertial force of the fluid element is equated to the sum of the external forces. The overall momentum equation in the x-direction is

$$\begin{aligned} &\frac{\rho}{g_c}\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z} + \frac{\partial u}{\partial t} \right) dxdydz \\ &= -\rho\frac{\partial \Omega}{\partial x} dxdydz + \left(\frac{\partial P_x}{\partial x} + \frac{\partial \tau_{yx}}{\partial y} + \frac{\partial \tau_{zx}}{\partial z} \right) dxdydz \end{aligned} \tag{24}$$

Similar expressions can be written for the y- and z-directions.

LAMINAR FLOW

In Section 2 calculation methods for handling flows through pipes and fittings will be presented. The frictional resistance of fluids as they flow through pipes and piping singularities (i.e., contractions, expansions, valves, tees, etc.), as well as the rates of heat and mass transfer between fluids and conduit walls, are critical parameters that establish design criteria for process operations. It is essential that we have as detailed an understanding as possible of how the fluid behaves in a flow situation. In this chapter we will examine briefly some of the predictions of the pipe flow equations developed from a momentum balance in the previous chapter. For the newcomer, we will also introduce the terms boundary layer and friction factor, and examine another common flow configuration given much analytical attention, namely, flow through an annulus.

VELOCITY AND SHEAR STRESS PROFILES

In the previous chapter an expression for the velocity profile of fluid flowing through a tube was derived

$$u = \frac{\left(P_1' - P_2'\right) R^2}{4\mu L}\left[1 - \left(\frac{r}{R}\right)^2\right] \tag{104}$$

The maximum point velocity can be obtained by setting r=0. Then

$$u_{\max} = \frac{\left(P_1' - P_2'\right) R^2}{4\mu L} \tag{108}$$

Or we can write this as

$$\frac{u_{\max}}{R^2} = \frac{1}{4\mu}\frac{\Delta P'}{L} \tag{109}$$

Hence, Equation 104 also can be written in the following form:

$$u = u_{\max}\left[1-\left(r/R\right)^2\right] \tag{110}$$

Earlier we defined the volumetric flowrate as Q=UA=U(π R^2), Where U is the average fluid velocity across the tube and A is the cross-sectional area of the tube. Or, perhaps more descriptive, we may write

$$dQ = 2\pi\ rudr \tag{111a}$$

and Q is obtained by integrating the point velocity over the cross section from $0 \leqslant r \leqslant R$.

$$Q = UA = \int_0^Q dQ = 2\int_0^R 2\pi rudr \tag{111b}$$

By substituting the value of U (Equation 110) into Equation 111b and performing the integration, the following relation is obtained:

$$U = \frac{U_{\max}}{2} \tag{112}$$

Hence, the velocity profile may be written as

$$\frac{u}{U} = 2\left[1-\left(r/R\right)^2\right] \tag{113}$$

Equation 113 is that of a parabola, and the profile is shown plotted in Figure 17(A).

Shearing stresses exist in the boundary layer of the flowing fluid and are exerted in the direction opposite to the flow. As noted earlier, shear stresses represent the force of resistance to flow. Again, from our force balance analysis in the preceding chapter, a linear relationship between the shear stress in the fluid and the distance from the tube axis was obtained (Equation 86, $\tau = \tau_W\left(r/R\right)$). This linear distribution is shown in Figure 17(B).

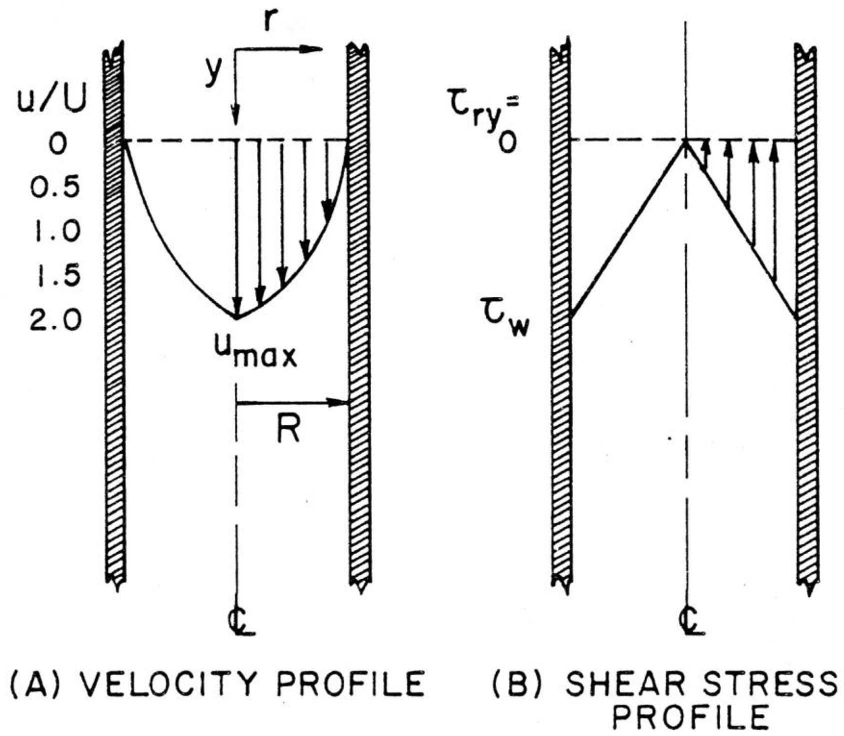


Figure 17. Velocity profile and shear stress distribution for laminar flow in a pipe.

FORMATION OF THE BOUNDARY LAYER

The boundary layer is defined by the region between the wall and a point in the viscous flowing fluid where the velocity is a maximum. The velocity gradient at the maximum velocity is zero.

Figure 18 illustrates how the boundary is formed. At the entrance of the conduit or pipe the boundary layer does not exist. As the flow continues along the pipe the boundary layer gradually increases in thickness until it eventually reaches a constant value at some distance from the entrance. Note that the dotted line in Figure 18 signifies the edge of the boundary layer. The last profile shown is that of the parabolic velocity distribution and signifies that fully developed flow exists.

RELATIONSHIP BETWEEN WALL SHEAR AND FRICTION FACTOR

Since shear stresses represent the force of resistance to flow, they are ultimately a measure of the friction losses. Friction losses in turbulent flow are known to be proportional to the kinetic energy of the fluid per unit volume and the area of the solid surface contacting the flow. The force per unit area, A_W, acting on the solid surface such as a pipe wall is

$$F / A_W \propto \frac{\rho U^2}{2g_c} = f \frac{\rho U^2}{2g_c} \tag{114}$$

where f is a proportionality constant.

We note that F/A_W is really the wall shear, τ_W; hence, $\tau_W = f\left(\rho U^2 / 2g_c\right)$.

In deriving the linear shear stress relation, it can be shown that wall shear may also be expressed as follows:

$$\tau_W = \left(\frac{R}{2}\right)\left(\frac{\Delta P}{L}\right) \tag{115}$$

By substituting for τ_W into the last expression, the proportionality constant f is solved. Factor f is commonly referred to as the friction factor, and Equation 116 defines the so-called “Fanning friction factor” for pipe flow

$$f = \frac{g_c R}{\rho U^2} \frac{\Delta P}{L} \tag{116a}$$

Or

$$f = \frac{g_c D}{2\rho U^2} \frac{\Delta P}{L} \tag{116b}$$

Where D is the pipe diameter.

For laminar flow in a tube, Poiseuille’s equation may be used to derive the following relation:

$$f = \frac{8\mu}{RU\rho} = \frac{16}{N_{\mathrm{Re}}} \tag{117}$$

N_{Re} is a dimensionless parameter called the Reynolds number. The friction factor definition in Equation 117 is good for Reynolds numbers up to 2000.

TURBULENCE AND TURBULENCE MEASUREMENT

This chapter provides introductory material to turbulence theory. The classical turbulent pipe flow models developed by Prandtl, von Karman and Deissler are rederived, and physical significance is attached to the various aspects of these analogies. Descriptive background on other types of turbulence encountered in process operations handling fluids is also included, along with a generous listing of primary references recommended for further reading. The last subsection in this chapter presents the operational theory behind anemometry techniques for turbulence and flow properties studies of flowing systems. A list of nearly 90 references has been compiled covering specific applications of anemometry techniques for various applications in turbulence investigations.

The laminar flow pattern described consists of a steady advance of fluid layers such that a streamline flow is maintained. If the fluid's flowrate is increased sufficiently this pattern is no longer maintained and the flow becomes unsteady. This type flow is characterized by chaotic movements of portions of the fluid in different directions superimposed on the main flow of the fluid; the phenomenon is referred to as turbulence. The complex movement of fluid elements within the flow can only be described in terms of time-averaged values. Figure 20(A) shows typical velocity fluctuations in turbulent flow as measured with a hot-wire anemometer at the center of a pipe for air flow.

Momentum transfers between adjacent regions of flowing fluid play a dominant role in establishing turbulence. These momentum transfers are inertial effects, which cause velocity and fluid density to take on important roles. By contrast, in laminar

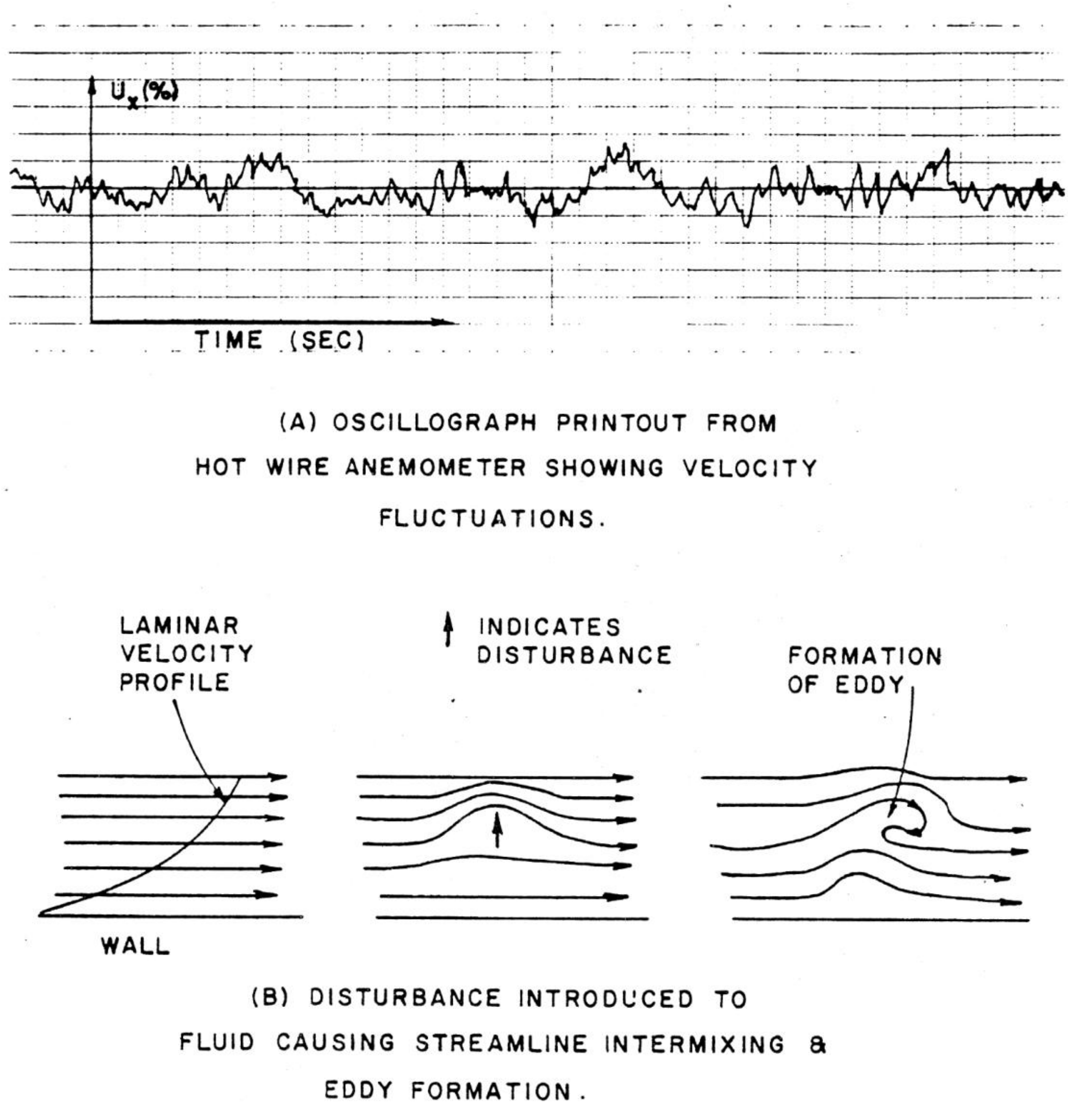


Figure 20. Shows typical turbulent fluctuations and the effect of random disturbance on laminar flow.

flow only purely viscous effects determine the nature of the flow. Turbulent flow dominates when inertial effects (denoted by ρu^2) become large in comparison in viscous forces (denoted by $\mu u/R$). As noted earlier, the change of conditions from laminar to turbulent flow is characterized by a dimensionless group called the Reynolds number:

$$N_{\mathrm{Re}} = \frac{\rho u D}{\mu} = \frac{uD}{\nu} \tag{122}$$

Where D is a characteristic length (for the case of simple pipe flow, the diameter) and ν is kinematic viscosity (μ/ρ), having dimensions of (length)^2/time.

The transition from laminar to turbulent flow for a small portion of flowing fluid is illustrated in Figure 20(B). The figure at the left shows a steady laminar flow situation. By introducing a random disturbance (middle figure), density and velocity are in part damped out by viscous forces. In fact, if we look back at Equation 122 we see it can be written in the form $N_{Re}=(\rho u)/(\mu u/D)$, which is the ratio of inertial to viscous forces. As more momentum is transferred into the disturbance, inertial forces eventually become much greater than the viscous term, and so-called eddy currents are formed (right sketch, Figure 20(B)).

Numerous investigators have studied the flow of fluids in pipes, confirming the flow regime dependency on Reynolds number. For flow in smooth pipes, the flow is always laminar for Reynolds number up to about 2000. Between Reynolds number of 2000 and 4000 the flow undergoes a transitional change (thus the name "transition region") from laminar to turbulent flow. Above Reynolds number of 4000 the flow is generally fully established turbulent flow.

The formation of turbulent eddy currents is best described by imagining ourselves to be moving within the fluid at the same velocity as the fluid's mean flowrate. If the flow is in a pipe, what we would see in the vicinity of the wall are patches of fluid torn into small eddies by the strong shearing forces within the fluid. These eddies tend to migrate toward the pipe center-line, some combining to form larger eddies, while many smaller ones simply diffusing into the “core” fluid with decaying intensity. We would further observe that these eddies are in fact superimposed on a much faster overall mean flow.

There are several situations that can lead to turbulent flow. Common cases are illustrated in Figure 21. One situation which we have just described is that of rapid flow of a fluid past a solid surface. This situation leads to unstable, self-amplifying velocity fluctuations, which form in the fluid in the vicinity of the wall and spread outward into the main fluid stream.

In a similar manner, turbulent eddies are formed from the velocity gradients established between a fast-moving fluid and a slower-moving fluid. A third general way in which turbulence is induced is by the relative movement of an object through the fluid streams. Examples of this last case are an impeller blade on an agitator and a falling sphere or cylinder. These situations cause eddies to form in the wake, resulting in an increase in the resistance of the movement of the object. This resistance is referred to as “form drag”.

In the case of stirred vessels, turbulence can be very intense near the tips of the rotor blades. Most of the turbulence throughout the vessel arises from velocity gradients, whereby portions of high-velocity fluid are thrown from impeller blades onto slower-moving fluid. Some of this turbulence is, however, attributed to the high shearing over the blades themselves, which form separation behind each blade or neighboring baffles.

PRINCIPAL FLOW RELATIONSHIPS

From boundary layer theory it can be shown that the mean velocity can be located at a distance of 37% of the total fluid level above the channel bottom. Velocity measurements obtained at this depth over a number of verticals can be averaged to obtain a mean velocity over the cross section. Another simple approach to obtaining the mean velocity in a vertical is to average measurements obtained at 0.2h and 0.8h. Where h is the height of the fluid surface above the channel bottom (referred to as the "bed").

From an engineering standpoint, flow capacity is of prime importance in open channels. Once a value for the mean fluid velocity has been determined for a given channel cross section, the discharge through that section can be established. Early work to establish engineering formulas was entirely empirical. The first relation developed for open channel flow was the Chezy formula. The basis for the Chezy formula is the following expression:

$$\text{Flow Resistance} = C'pLU^2 \tag{258}$$

Where C' = friction coefficient

P =length of the wetted perimeter of the channel

L = length of the channel

If the channel is inclined at an angle, θ , from the horizontal, then the slope, S, of the fluid level is $\sin\theta = h/L$, where h denotes the level over a finite fluid element length, L. Figure 189 defines the system under consideration.

Defining W as the weight of a finite volume of fluid designated between points 1 and 2, then

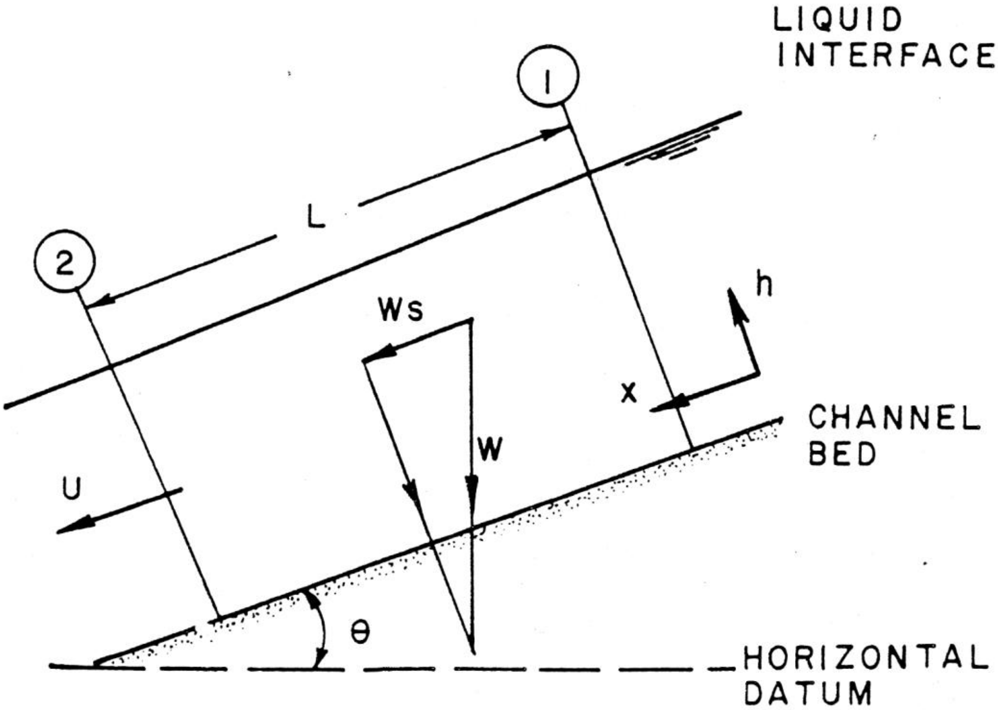


Figure 189. Defines terms for a sloping channel used in the derivation of the Chezy formula.

$$W_S = ALWS \qquad (259)$$

Where A is the cross-sectional area of the channel flow and W is the specific weight of the fluid.

WS is actually a force causing flow, hence, equating Equations 258 and 259, and, rearranging,

$$U^2 = \frac{W}{K}\frac{A}{P}S \qquad (260)$$

Equation 260 is the Chezy formula, which is more commonly written as

$$u = C\sqrt{h_D S} \qquad (261)$$

Where h_D is the hydraulic mean depth, h_D =A/p, and C is the Chezy constant, $C = \sqrt{W/K}$.

Chezy's constant can be evaluated from the following formula:

$$C = \frac{23 + \frac{0.00155}{S} + \frac{1}{\varepsilon}}{1 + \left(23 + \frac{0.00155}{S}\right)\frac{\varepsilon}{\sqrt{h_D}}} \tag{262}$$

Where ε is a coefficient of roughness for the channel material.

A second flow formula heavily used for open channel flows was already presented back in Chapter 2.2-the Manning equation (Equations 48 and 49). The Manning equation is restated here in notation more consistent with our present discussion:

$$u = \frac{1}{\varepsilon} h_D^{\frac{2}{3}} S^{\frac{1}{2}} \quad \text{in SI units} \tag{263a}$$

Or

$$u = \frac{1.49}{\varepsilon} h_D^{\frac{2}{3}} S^{\frac{1}{2}} \quad \text{in British units} \tag{263b}$$

Of the two relationships, the Manning formula is to be preferred for accuracy.

Unlike full pipe flow, the roughness element, ε , for open channel flows is sensitive to the hydraulic state of the channel. Usually, assigning a reliable value for ε will depend on experience and judgment. Table 41 gives a typical range of values for ε for different straight channel conditions. For nonstraight channels, ε values should be increased by 30%.

The simple flow expressions presented in this subsection form the starting basis for describing open channel flow and channel discharge. However, the complexity of analysis in describing the characteristics of an open channel flow situation depends on the nature of the flow. There are three general cases that are frequently encountered in engineering; these are uniform flow in channels, nonuniform flows and unsteady flows in open channels.

PUMPS AND PUMPING SERVICES

This chapter presents general information and procedures for specifying pumps and process plant pumping services. Details on pump application technology are presented in subsequent discussions in this chapter. This chapter only provides supplemental information to acquaint the plant engineer with pumping technology and nomenclature. Emphasis is given to centrifugal pumps as these are most widely encountered in the chemical process industries. For more detailed discussions on pump types, designs and performance, the reader is referred to the literature [12, 13]. Note also that data presented here represent typical conditions and should therefore only be applied for approximations. The manufacturer's actual data should always be used when specifying a specific pump style for an intended service.

PUMP CLASSIFICATIONS

The pump types most often employed in process plant applications fall into the following classes: centrifugal, axial, regenerative turbine type, reciprocating, metering and rotary. There are two categories of pumps under which these classes are grouped: dynamic pumps and positive displacement pumps.

Dynamic pumps include the classes centrifugal and axial. These pumps operate by developing a high liquid velocity and converting the velocity to pressure in a diffusing flow passage. In general, they tend to have lower efficiency than positive displacement pumps. However, they do operate at relatively high speeds to permit high flow rate in relation to the physical size of the pump. Also, dynamic pumps usually have significantly lower maintenance requirements than positive displacement pumps.

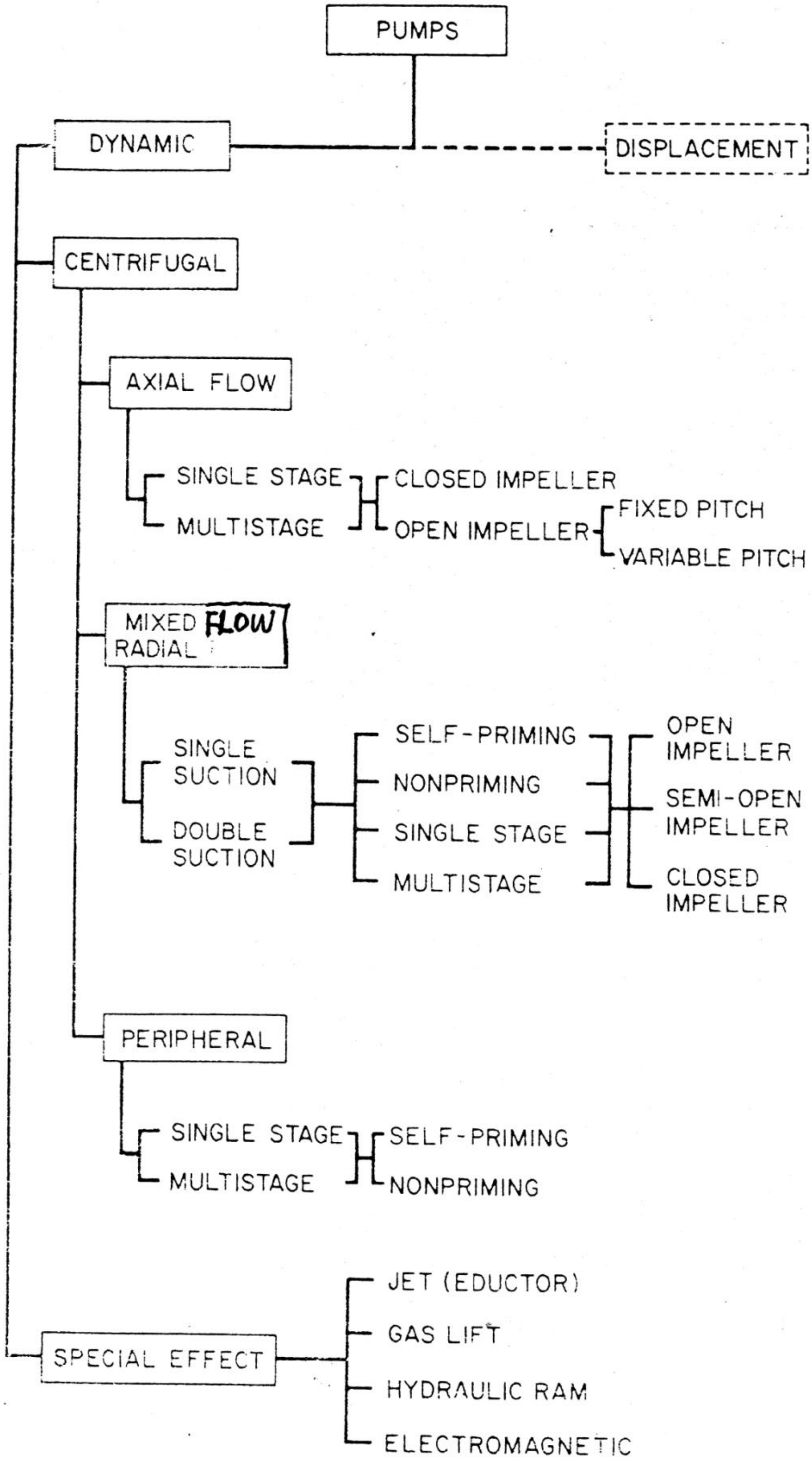


Fig. 1 Classification of dynamic pumps.

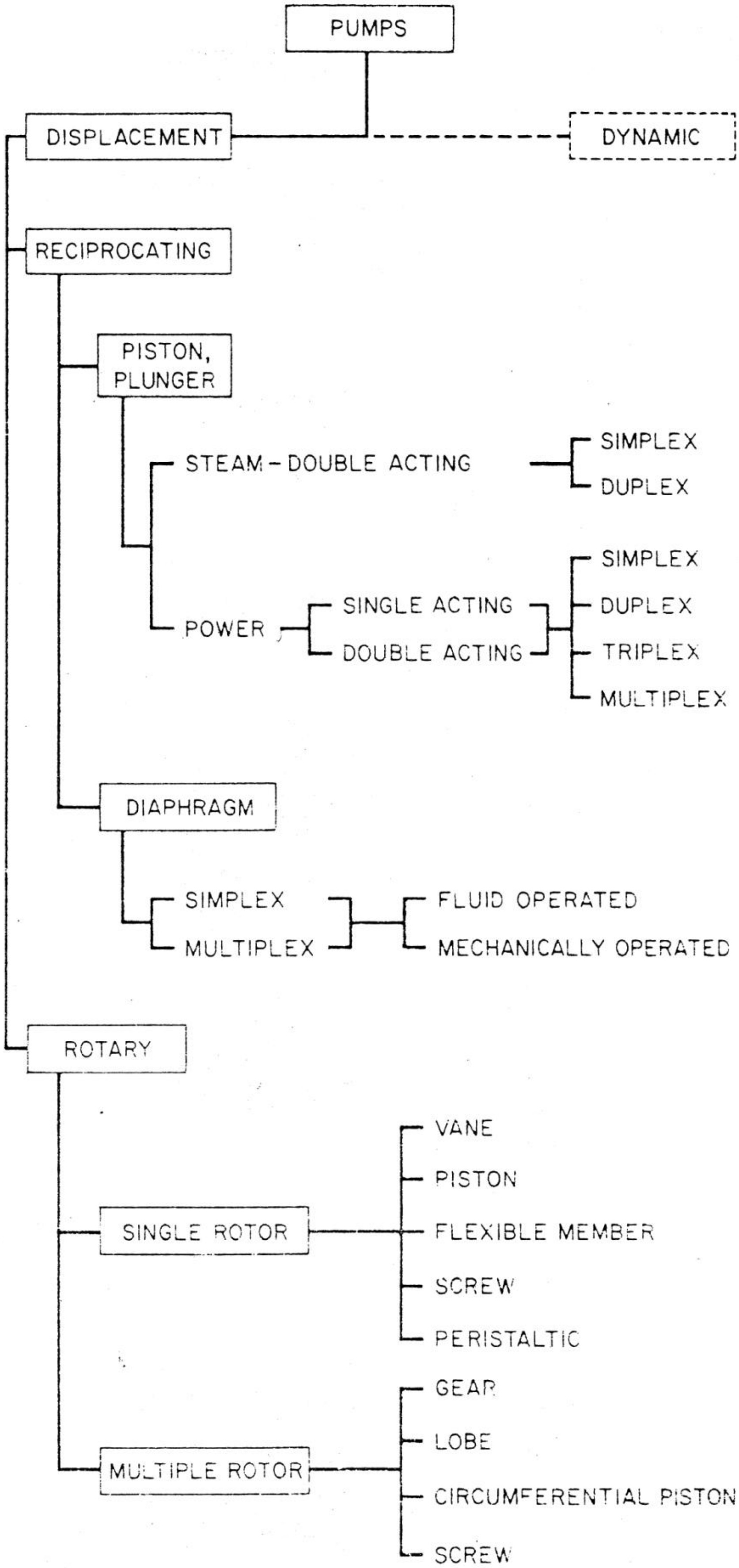


Fig. 2 Classification of displacement pumps.

Positive displacement pumps operate by forcing a fixed volume of fluid from the inlet pressure section of the pump into the discharge zone of the pump. With reciprocating pumps this is performed intermittently. In the case of rotary screw and gear pumps it is done continuously. This category of pumps operates at lower rotative speeds than dynamic pumps. Positive displacement pumps also tend to be physically larger than equal-capacity dynamic pumps.

CENTRIFUGAL PUMPS

Centrifugal pumps accomplish the generation of pressure by the conversion of velocity head into static head. The rotary motion of impellers adds energy to the service fluid in the form of a velocity increase. This velocity increase is converted into static head in the diffusing section of the casing. Centrifugal pumps have no valves. The flow is uniform and free of low-frequency pulsations. A pump operating at a fixed speed will develop the same theoretical head in feet of flowing fluid, regardless of density. However, the pressure corresponding to the developed head (in psi) depends on the fluid density.

The parameters that establish the maximum head (in feet of fluid) that a centrifugal pump can develop are the pump speed (rps), impeller diameter and the number of impellers in series combination. Impeller design and blade angle mainly affect the slope and shape of the head-capacity curve and normally have minor effects on the developed head. Conventional centrifugal pump impellers have a maximum tip speed around 200 fps (60 meter/s). In slurry pumps, impellers are limited to about half this tip speed to limit erosion problems.

Normal and maximum viscosity ranges are a major consideration in pump selection because of possible deterioration in performance with increasing viscosity. Deterioration can be both continuous and gradual. Table 20 can serve as a guide in selecting the proper centrifugal pump type for an application.

Most centrifugal pump types are not self-priming, meaning that they are not capable of evacuating vapor from the suction line so that liquid can flow into the line and into the pump casing without external assistance. The impellers on centrifugal pumps are designed specifically for efficient liquid pumping and are not operated at high enough tip speeds to convert them into vapor compressors. The differential head that the pump impeller can deliver is the same on vapor as on liquid. However, the equivalent differential pressure rise capability is much lower with vapor. Thus, centrifugal pump impellers are not capable of generating a significant reduction in the pressure exerted by the vapor in the suction line to permit liquid flow.

To prime a centrifugal pump before starting, the suction line and pump casing must be filled with liquid. When the suction source is at positive pressure or is positioned above the pump, priming is done by opening the suction valve and releasing (venting) the trapped vapor from a valved connection on the pump casing or discharge line (located inside the discharge block valve). Liquid then will flow into the suction line and pump casing to displace the venting vapor.

Most centrifugal pumps have a self-venting feature. Small amounts of vapor trapped within the casing at startup (after suction priming is complete) are swept out into the discharge line when the pump is started. Horizontally split casings are not arranged to be self-venting, however, and are equipped with specially designed valved vent connections requiring manual operation. Single-stage centrifugal pumps with top discharge connections have good self-venting performance, even though the casing shape places a small high point vapor pocket in the top of the discharge volute.

A brief description of the major types of centrifugal pumps is given below.

Single-Stage Overhung

This design has a single-stage overhung impeller. Its casing is supported at the centerline. Two shaft bearings are mounted close together in the same bearing bracket, with the impeller cantilevered or overhung beyond them. This design is illustrated in Figure 33. This type design usually has top suction and discharge flanges; wearing rings both on the front and back of the impeller and casing; a single suction, closed impeller; and a single stuffing box fitted with a mechanical seal. There are also water cooling options on the pedestal, stuffing box and bearings. This pump type is well suited for high-temperature operation and can be used for handling flammable liquids.

Two-Stage Overhung

This is simply a modified version of the single-stage process pump. It is capable of higher head than its single-stage counterpart. In this style, the stuffing box pressure is roughly halfway between suction and discharge pressures.

Single-Stage (Impeller-Between-Bearings)

These designs have their impellers mounted between the bearings and thus have two stuffing boxes. Single-stage versions are capable of developing heads up to 1080 feet (330 meters). Casings can be axially split for temperatures up to 500℉

(260℃) and radially split for temperatures up to 850℉ (455℃).

Inline Pumps

These are vertical pumps with casings designed to be bolted directly to piping (similar to a valve). There are two basic configurations: coupled and close-coupled. Service life and maintenance requirements for both styles are about the same. Generally, this type of pump is preferred if the cost of piping associated with installation can be reduced over the conventional horizontal styles. Chemical industry standards for inline pumps are summarized by Vetter and Fritsch [14].

High-Speed Centrifugal Pumps

These pumps are single-impeller models designed for speeds typically in the range of 170 to 280 rps and as high as 400rps. They are capable of heads up 5300 feet (1600 meters). Pumping temperature are limited to about 500℉ (260℃). High-speed pumps tend to have high NPSH requirement due to the sudden velocity increase as the liquid enters the impeller. Maintenance requirements for these pumps tend to be higher than for conventional speed, single-stage pumps; however, it is about the same as multistage models, with which they compete for high head services.

Chemical Pumps

This class of pumps is designed with casing shapes that are cast in high-cost alloys. Casings are more frequently foot supported or bearing bracket supported than centreline supported. These pumps are limited to relatively low temperature, pressure and flowrates.

Slurry Pumps

Slurry pumps are used in services that have severe conditions of slurry pumping. They have a number of special design features that make them well-suited for this type service:

1. Wide flow passages to avoid clogging;
2. Open/semiopen impellers, which are less sensitive than closed impellers to clogging;
3. Arrangements that break up large particles;
4. Low fluid velocities, as low rotating and peripheral speed are possible;
5. Adjustable rotor position to restore axial clearance without dismantling the pump; and
6. Replaceable wearing plates and pumping plates on back of impeller, instead of wearing rings, which are subject to erosion.

Canned Pumps

These are motor pump units with the rotating rotor and impeller housed entirely within a pressure casing. This type design eliminates the need for a stuffing box. The pumped fluid serves both as a lubricant for bearings and as a coolant for the motor. Designs are limited to low-flow, low-pressure and low-temperature service.

Horizontal and Vertical Multistage Pumps

Horizontal multistage pumps are limited to about 12 stages due to the difficulty in limiting deflection over the long span between bearings. They have NPSH requirements that match those of single-stage pumps of the same capacity. These pumps are well suited for corrosive fluids handling.

Vertical multistage pumps may have as many as 24 (sometimes more) stages.

Some high differential pressure models utilize opposed thrust arrangements. Below roughly 1200 feet (370 meters) head, pumps for as low as 1 foot (0.3 meter) NPSH at the suction flange are available. High specific speed impellers are frequently used. The first stage is normally at the bottom of the assembly, below grade. Vertical multistage pumps need a large number of close running clearances. Hence, these pumps are sensitive to damage by solids ingestion and by dry or two-phase generating conditions. In general, maintenance requirements for vertical multistage pumps are greater than for their horizontal counterparts.

Regenerative Turbine Pumps

Some manufacturers prefer to place regenerative turbine pumps in a separate class. In general, this type pump greatly resembles a conventional centrifugal pump but has the distinction of a much steeper head-capacity curve. The impeller consists of a solid disc with fluted vanes on each side of the perimeter, which impart energy to the liquid by multiple excursions from the impeller to the stator and back to the impeller (tracing dual screw shaped paths along the stator annulus).

Turbine pumps have shutoff pressures typically two or three times the design level. The steepness of the head curve causes the power requirement curve to rise as flow decreases, thus peaking at shutoff. Drivers for turbine pumps must be sized for minimum flow, rather than normal flow. Usually, a safety valve is needed inside the discharge block valve.

This type pump is extremely sensitive to dirt, temperature shocks, and to piping forces and moments on the pump flanges.

POSITIVE DISPLACEMENT PUMPS

Positive displacement pumps operate by forcing a fixed volume of liquid from the inlet pressure zone of the pump into the discharge zone of the pump. A brief description of each of the major types of positive displacement pumps is given below.

Reciprocating Pumps

Reciprocating pumps produce pulsating flow, develop high shutoff or stalling pressure, display constant capacity when motor driven, and are subject to vapour binding at low NPSH conditions. There are several construction styles. One type, the direct acting steam pump, consists of a steam cylinder end in line with a liquid cylinder end, with a straight rod connection between the steam piston and the pump piston or plunger.

Direct-acting steam pumps are available as simplex (one steam and liquid cylinder) and duplex (dual side by side) units. Duplex units are employed in larger capacity services and to reduce the flow pulsations below that of the simplex. Dual pumps are designed with an interconnecting steam valve linkage arrangement so that one side pumps when the other side reaches the end of its stroke. Steam pumps consist of rod and piston design and are double acting (that is, each side pumps on every stroke). Consequently, a duplex pump will have four pumping strokes per cycle.

Another construction style is the power pump. Power pumps convert rotary motion to low-speed reciprocating motion via speed reduction gearing, a crankshaft, connecting rods and crossheads. Plungers or pistons are driven by the crosshead drives. Rod and piston construction, similar to duplex double-acting steam pumps, are used by the liquid ends of the low-pressure, higher-capacity units. The higher-pressure units are normally single-acting plungers. This latter style generally employs three (triplex) plungers. Three or more plungers substantially reduce flow pulsation relative to simplex and even duplex pumps.

In general, power pumps have high efficiency and are capable of developing very high pressures. They can be driven either by electric motors or turbines. They are relatively expensive pumps and rarely can be justified on the basis of efficiency over centrifugal pumps. However, they are frequently justified over steam reciprocating pumps where continuous-duty service is needed due to the high steam requirements of direct-acting steam pumps.

In general, the effective flowrate of reciprocating pumps decreases as viscosity increases because the speed must be reduced. High viscosity also leads to a reduction in pump efficiency. By contrast to centrifugal pumps, the differential pressure generated by reciprocating pumps is independent of fluid density. It is entirely dependent on the amount of force exerted on the piston.

Reciprocating pumps are most often used for sludge and slurry services, particularly where other types are inoperable or troublesome. Maintenance in such services tends to be high because of valve, cylinder, rod and packing wear.

Metering Pumps

Metering pumps are positive displacement pumps that provide precision control of very low flowrates. Flowrates can range from 1.6×10^{-3} to 0.16 gpm; However, there are some higher-capacity models that provide flows up to 0.66 gpm. Flow accuracy is typically within ±1%. Control schemes for metering pumps are used for controlling the proportioning of additives injected into the main flow stream. Other names for these type pumps are “proportioning pumps” and “controlled volume” pumps.

Metering pumps are available in two construction styles: diaphragm and packed plunger. The diaphragm design employs a hydraulic oil barrier between the reciprocating plunger and an impervious diaphragm, which, in turn, contacts the pumped liquid. The stuffing box works in lube oil in this design, so there is no process liquid leakage. With the packed plunger arrangement, the design resembles a small version of a conventional larger plunger pump, in which the stuffing box is exposed to the service liquid.

Usually the driver consists of an electric motor. Basically, the same design criteria applied to larger motor-driven reciprocating pumps can be applied to proportioning pumps.

Capacity variations normally are handled by manual resetting of the stroke adjustment. Controls are available for automatic stroke resetting and for remote, manual stroke resetting. The typical efficiency of this type pump is around 20%. Generally, viscosity effects on power requirements are negligible.

Finally, it should be noted that these pumps are designed for clean service. Nozzle connections and valves of metering pumps are small and, thus, are subject to plugging and/or valve sticking when handling dirty liquids. Figure 35 shows a cutaway view of a metering pump.

Diaphragm Pumps

This type of positive displacement pump operates by the periodic movement of a flexible diaphragm. It has the advantages of no stuffing boxes and high tolerance to abrasive slurries.

The diaphragm is flexed by pulsating fluid pressure on the drive side. Compressed air normally is used; however, steam and hydraulic oil systems are also available. Drive pressures normally pulsate between 0 and 15 psi (1-105 kPa) above the average discharge pressure level in the process stream.

Rotary Pumps

There are a wide variety of rotary pumps on the market; however, liquid services are limited primarily to external gear pumps and screw pumps. Sliding vane and internal gear pumps find limited application to process plant services.

The single screw pump is a special type of screw pump for handling slurries with relatively large particles. This design allows little fracturing of particles and very little abrasion damage to the pump. Single-screw pumps are employed extensively in the food processing and chemical industries for handling solid/liquid mixtures that are either abrasive or require gentle handling of the solid particles.

The working principle of the screw pump is that of Archimede’s screw, invented by the Greek mathematician about 200 BC. Screw pump applications include fuel, lube and crude oil service; navy and marine cargo; oil burners; slurry handling; and a variety of high-viscosity materials such as polymers, copolymers and elastomers, syrups, fats and grease, soaps and solvents.

Today, both single-rotor and multiple-rotor screw pumps are commercially available. Pumping action is accomplished by progressing cavities, which advance along the rotating screw from inlet to outlet. This axial flow pattern minimizes vibration, producing smooth flow.

In the twin-screw pump, two sets of screws rotate and mesh in an accurately bored casing, having rather tight operating clearances maintained between them. The mechanical displacement of fluid from inlet to outlet is generated by trapping a slug of fluid in the helical cavity, created by the meshing of the screws.

In most twin screw designs, the clearance between screws is maintained by a pair of timing gears mounted on the shafts. These gears also transmit power from the drive shaft to the driven shaft. The body consists of a casing with two precision-machined bores, which house the rotating screws. Fluid passes from the inlet chamber into the pumping chamber and then to the discharge chamber. Screw pump bodies are commonly made of cast iron, ductile iron, cast steel or 316 stainless, depending on such factors as pressure requirement, need for corrosion or galling resistance, pumping temperature, etc. Body bores sometimes are chrome plated to improve surface finish, antigalling characteristics and hardness.

The screws are designed to withstand hydraulic pressure within the screw channels, which exerts both a radial and an axial force on the screws. Axial forces are balanced by using the matched pair of screws on each shaft. Radial forces tend to cause shaft deflection, and are resisted by the fluid film trapped between the screws.

Screws and shafts may be of “pinned” or “integral” design. In the pinned screw design, the screws are machined as separate pieces and mounted on the shaft with pins or keys. This design is used for relatively low pressures, up to about 500 psi.

The integral screw arrangement, with screws and shaft machined from a single forging or piece of barstock, provides much higher pressure, viscosity and shaft torque capabilities.

Timing gears are used to transmit power from one rotor shaft to the other and to maintain the proper clearance between the pumping screws. Most screw pumps are designed with relatively small internal clearances between the pumping screws. This mandates high-precision timing gears if metal-to-metal contact is to be avoided. The positioning of timing gears relative to the pumping screws is called "the timing of the screws." Timing gears may be double-helical type, which maintain the proper angular and axial relationship between the pumping screws. Single-helical or spur designs also are used. The latter maintain only the angular relationship, relying on the thrust bearing to maintain the proper axial relationships between pumping screws.

Some screw pump designs work without timing gears. In these designs, there is a drive screw that is used to drive the driven screws. The design of twin-screw pump bodies and the arrangement of screw assemblies are interrelated. Figure 37 illustrates typical body designs.

There are a variety of novel rotary pump types commercially available. The more conventional styles are compared in Table 21.

The main reason for selecting rotary pumps over centrifugals is to take advantage of their high viscosity capability. In addition, rotaries are simple in design and efficient in handling flow conditions that are generally considered too low for economic application of centrifugals. The importance of viscosity in rotary pump design is summarized in Table 22. Rotary pumps designed to handle high-viscosity liquids must be operated at reduced speeds (and, thus, at reduced flowrates).

Note that proper pump selection is not only dependent on the fluid viscosity level, but also on the rheological properties of liquid. For non-Newtonian fluid behavior, pump design and selection will depend also on how viscosity changes with shear rate. Shear rate/shear stress data should be included when preparing design specifications.

CHARACTERISTICS OF CENTRIFUGAL PUMPS

The selection of a centrifugal pump for an energy-efficient pumping system requires an understanding of the principles of mechanics and physics that can affect the pumping system and the pumped liquid. The efficiency of a centrifugal pump is also dependent on the behavior of the liquids being pumped. The principles of centrifugal pump operation that govern head and flow must be understood clearly before pump performance can be evaluated accurately.

The behavior of a fluid depends on its state-liquid or gas. Liquids and gases offer little resistance to changes in form. Typically, fluids such as water and air have no permanent shape and readily flow to take the shape of the containing enclosure when even a slight shear loading is imposed. Factors affecting behavior of fluids include:

1. Viscosity
2. Specific gravity
3. Vapor pressure

In Section 1 it was noted that viscosity is the resistance of a fluid to shear motion-its internal friction. The molecules of a liquid have an attraction for each other. They resist movement and repositioning relative to each other. This resistance to flow is expressed as the viscosity of the liquid. Dynamic viscosity also can be defined as the ratio of shearing stress to the rate of deformation. The viscosity of a liquid varies directly with temperature; therefore, viscosity is always stated at a specific temperature.

Liquid *viscosity* is very important in analyzing the movement of liquids through pumps, piping and valves. A change in viscosity alters liquid handling characteristics in a system; more or less energy may be required to perform the same amount of work. In a centrifugal pump, an increase in viscosity reduces the pressure energy (head) produced while increasing the rate of energy input. In piping system, a liquid with a high viscosity has a high energy gradient against which a pump must work, and more power is required than for pumping low-viscosity liquids.

Specific gravity is the ratio of the density of substance to that of a reference substance at a specified temperature. Water at 4℃ is used as the reference for solids and liquids. Air generally is used as the reference for gases. The specific gravity of a liquid affects the input energy requirements, or brake horsepower (bhp), of centrifugal pumps. Brake horsepower varies directly with the specific gravity of the liquid pumped. For example, water at 4℃ has a specific gravity of 1.0. Table 23 includes some specific gravity values for water at selected temperatures.

Specific gravity affects the liquid mass but not the head developed by a centrifugal pump, as shown in Table 24. The specific gravity also affects the energy required to move the liquid and, therefore, must be used in determining the pump’s horsepower requirement.

Vapor pressure is the pressure at which a pure liquid can exist in equilibrium with its vapor at a specified temperature. Fluids at temperatures greater than their specified (critical) temperature will exist as single-phase liquids (vapors), with no distinction between gas and liquid phases. At less than the critical temperature, two fluid phases can coexist; the denser fluid phase exists as a liquid and the less dense phase as a vapor. At a specific temperature, the liquid phase is stable at pressures exceeding the vapor pressure and the gas phase is stable at pressures less than the vapor pressure.

For a fluid to exist in a liquid state its surface pressure must be equal to, or greater than, the vapor pressure at the prevailing temperature. For example, water has a vapor pressure of 0.1781 psia at 10℃ and 14.69 psia at 100℃. The vapor pressure of a volatile liquid (such as ether, alcohol or propane) is considerably higher than that of water at the same temperature; consequently, much higher pressures must be applied to maintain volatile materials in their liquid states. The surface pressure of a liquid must be greater than its vapor pressure for satisfactory operation of a centrifugal pump.

Centrifugal pump characteristics remain constant unless an outside influence causes a change in operating conditions. Three conditions can alter pump performance:

1. changes in impeller or casing geometry;
2. increased internal pumping losses caused by wear; and
3. Variation of liquid properties.

For example, if the impeller passages become impacted with debris, the head-flow relationship will be reduced. Similarly, performance will decline if mechanical wear increases the clearance between the rotating and stationary parts of the pump.

Except for specially designed pumps, most centrifugal pumps can handle liquids containing approximately 3~4% of gas (by volume) without an adverse effect on performance. An excess of gas will reduce the flow of liquid through the pump and, under certain conditions, flow will cease, setting up a condition that may damage the pump.

The function of a pump is to move liquids by imparting pressure energy (head) to the liquid. The ability of the pump to perform its function is based on the Bernoulli theorem, which states that energy cannot be created or destroyed, but can only be converted in form. A pump converts mechanical energy into pressure energy. Part of the converted energy is required to overcome inertia and move the liquid; most of the remaining energy is stored in the liquid as elevated pressure, which can be used to perform useful work outside the pump. A centrifugal pump is basically a velocity machine designed around its impeller. The interaction between the impeller and its casing produces the characteristics of head or pressure energy.

To review, the *developed head* is a function of the difference in velocity between the impeller vane diameter at entrance and the impeller vane diameter at exit. The expression of theoretical head can be related to the law of a falling body:

$$H = \frac{u^2}{2g}$$

Where H = height or head (ft)

u = velocity of moving body (fps)

g = acceleration of gravity (32.2 ft/s^2)

When the height of fall is known (for example, H = 100 ft), the terminal velocity can be determined (in this case u=80.3 fps). Conversely, if the direction of motion is reversed, a liquid exiting through an impeller vane tip at a velocity of 80.3 fps reaches a velocity of 0 fps at 100 ft above the impeller tip (Figure 42).

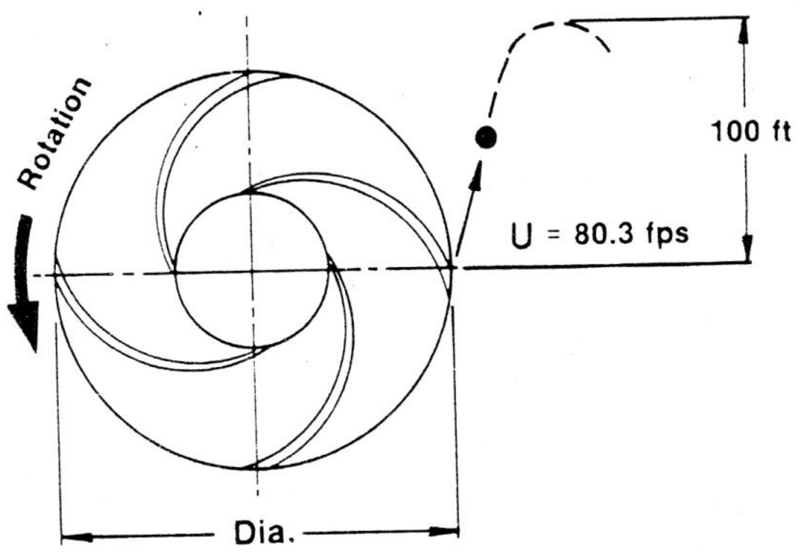


Figure 42. Example to determine theoretical impeller diameter.

When the developed head required is known, the theoretical impeller diameter for any pump at any rotational speed can be determined from the equation for peripheral velocity of a round rotating body:

$$D = \frac{229.2u}{N} \tag{90}$$

Where D = unknown impeller diameter (in.)

u = velocity (derived from $u = \sqrt{2gH}$) (fps)

N = rotational speed of the pump (rpm)

If pump head is 100 ft and rotational speed is 1750 rpm, the formula indicates that a 10.52-in. theoretical diameter impeller is required.

The function of a centrifugal pump is to transport liquids by transferring and converting mechanical energy (foot-pounds of torque) from a rotating impeller into pressure energy (head). By the transfer of this energy to a liquid, the liquid can perform work, move through pipes and fittings, rise to a higher elevation or increase the pressure level. Because a centrifugal pump is a velocity machine, the amount of mechanical energy per unit of fluid weight transferred to the liquid depends on the peripheral velocity of the impeller, regardless of fluid density. This energy per unit of weight is defined as pump head and is expressed in feet of liquid. If the effect of liquid viscosity is ignored, the head developed by a given pump impeller at a given speed and flowrate remains constant for all liquids.

Head is the vertical height of a column of liquid. The pressure this liquid column exerts on the base surface depends on the specific gravity of the liquid. A 10 ft column of liquid with a specific gravity of 0.5 exerts only 2.16 psi; a 10 ft column of liquid with a specific gravity of 1.0 exerts 4.33 psi.

The formula for converting feet of head to pressure is

$$H = \frac{P \times 2.31}{Sg} \tag{91}$$

Where P = pressure (psi)

Sg = specific gravity

The proper selection of a centrifugal pump requires that pressure be converted to feet of head. If not, a pump that is incapable of imparting the required energy to a liquid may be installed before the inadequacy is discovered. The heads required to produce a pressure of 100 psi for three liquids of different specific gravities are given in Table 25.

Table 25. Heads Required to Produce 100 psi Pressure for Different Liquids

Head equivalent of 100 psi for liquids of different specific gravity	
Specific Gravity	Head (ft of liquid)
0.75	308
1.0	231
1.2	193

From this analysis a single-stage pump selected for pumping a liquid with a specific gravity of 1.2 will have the lowest peripheral velocity at the impeller tip, while the pump for a liquid with 0.75 specific gravity will have the highest peripheral velocity.

The volumetric flowrate is a function of the peripheral velocity of the impeller and the cross-sectional areas in the impeller and its casing; the larger the passage area, the greater the flowrate. Consequently, the physical size of the increases with higher flow requirements for a given operating speed. The liquid flowrate is directly proportional to the area of the pump passages and can be expressed as

$$Q = \frac{u \times A}{0.321} \qquad (92)$$

Where Q = flowrate (gpm)

u = velocity (fps)

A = area (in.^{2})

For example, a centrifugal pump having a 15 fps liquid velocity at the discharge flange can pump approximately 330 gpm through a 3-in.-diameter opening. If the flow requirements are increased to approximately 1300 gpm and the velocity remains at 15 fps, a 6-in. -diameter opening will be required. Liquid flow is also directly proportional to the rotational speed that produces the velocity. The four preceding equations establish two relationships:

1. Head is directly proportional to the square of the liquid velocity.
2. Flow is directly proportional to the peripheral velocity of the impeller.

Cavitation in a centrifugal pump can be a serious problem. Liquid pressure is reduced as the liquid flows from the inlet of the pump to the entrance to the impeller vanes. If this pressure drop reduces the absolute pressure on the liquid to a value equal to or less than its vapor pressure, the liquid will change to a gas and form vapor bubbles. The vapor bubbles will collapse when the fluid enters the high-pressure zones of the impeller passages.

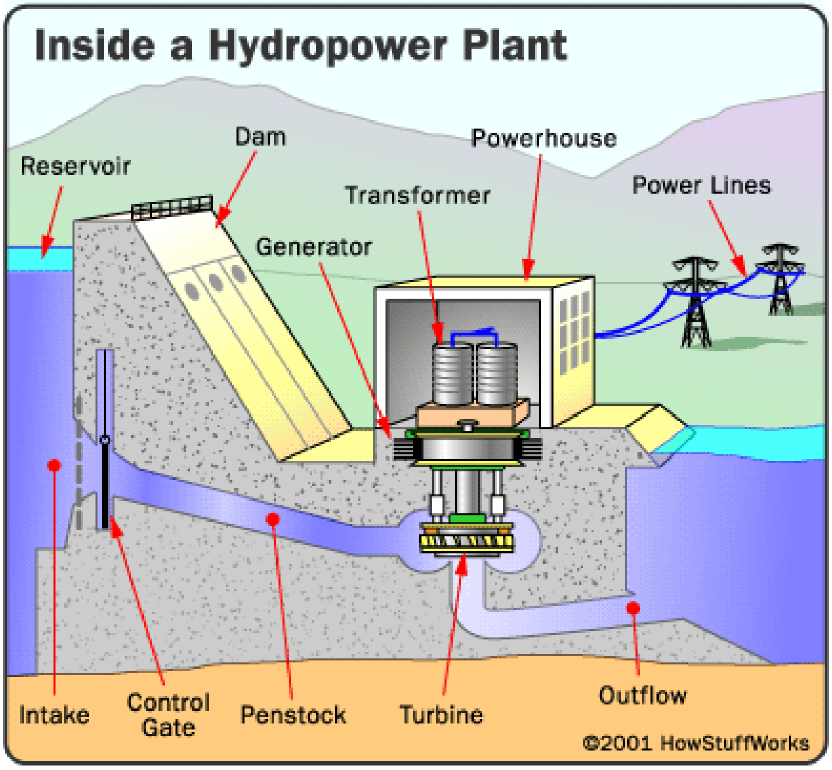
This collapse is called cavitation and results in a concentrated transfer of energy, which creates local forces. These high-energy forces can destroy metal surfaces; very brittle materials are subject to the greatest damage. In addition to causing severe mechanical damage, cavitation also causes a loss of head and reduces pump efficiency. Cavitation will also produce noise.

If cavitation is to be prevented, a centrifugal pump must be provided with liquid under an absolute pressure that exceeds the combined vapor pressure and friction loss of the liquid between the inlet of the pump and the entrance to the impeller vanes. Chapter 2.8 discusses cavitation more completely.

Hydraulic Turbines

Introduction

A water/hydraulic turbine is a rotary engine that takes energy from moving water. Water turbines were developed in the nineteenth century and were widely used for industrial power prior to electrical grids. Now they are mostly used for electric power generation. They harness a clean and renewable energy source.



Water wheels have been used for thousands of years for industrial power. Their main shortcoming is size, which limits the flow rate and head that can be harnessed. The migration from water wheels to modern turbines took about one hundred years. Development occurred during the Industrial revolution, using scientific principles and methods. They also made extensive use of new materials and manufacturing methods developed at the time.

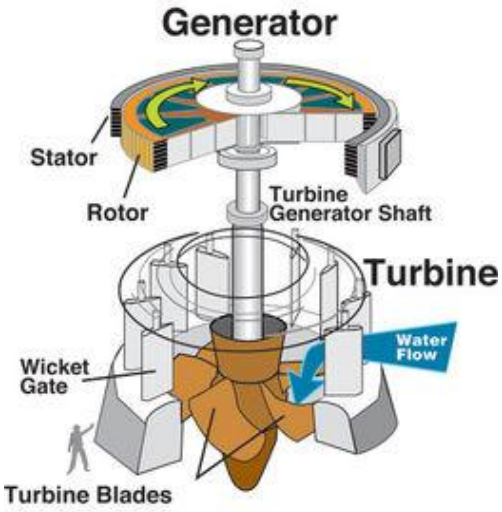
Water wheel

The word turbine was introduced by the French engineer Claude Bourdin in the early 19th century. The main difference between early water turbines and water wheels is a swirl component of the water which passes energy to a spinning rotor. This additional component of motion allowed the turbine to be smaller than a water wheel of the same power. They could process more water by spinning faster and could harness much greater heads.

Theory of operation

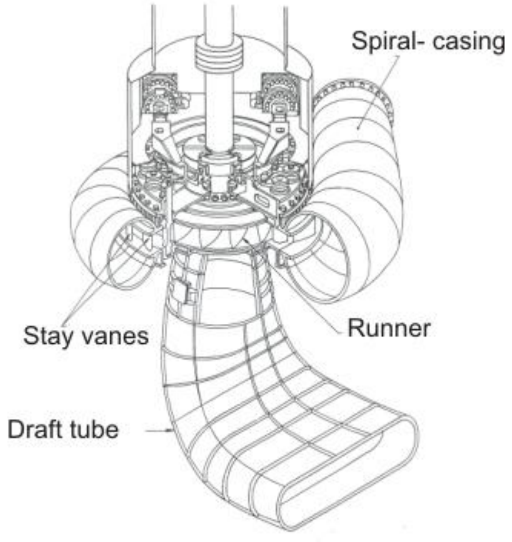
Flowing water is directed on to the blades of a turbine runner, creating a force on the blades. Since the runner is spinning, the force acts through a distance (force acting through a distance is the definition of work). In this way, energy is transferred from the water flow to the turbine.

The precise shape of water turbine blades is a function of the supply pressure of water, and the type of impeller selected.

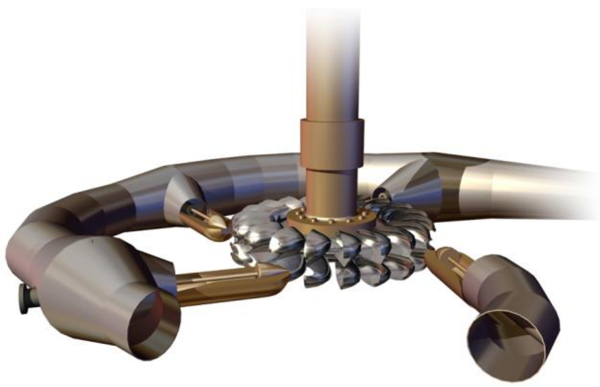


Types of hydraulic turbomachine

Water turbines are divided into two groups: reaction turbines and impulse turbines.



Reaction turbines



Impulse turbines

Reaction turbines

Reaction turbines are acted on by water, which changes pressure as it moves through the turbine and gives up its energy. They must be encased to contain the water pressure (or suction), or they must be fully submerged in the water flow. Newton's third law describes the transfer of energy for reaction turbines. Most water turbines in use are reaction turbines and are used in low (<30m/98ft) and medium

(30-300m/98-984ft) head applications. In reaction turbine, pressure drop occurs in both fixed and moving blades.

Reaction turbines: Bulb turbine, Kaplan turbine, Francis turbine and Pump turbine.

Impulse turbines

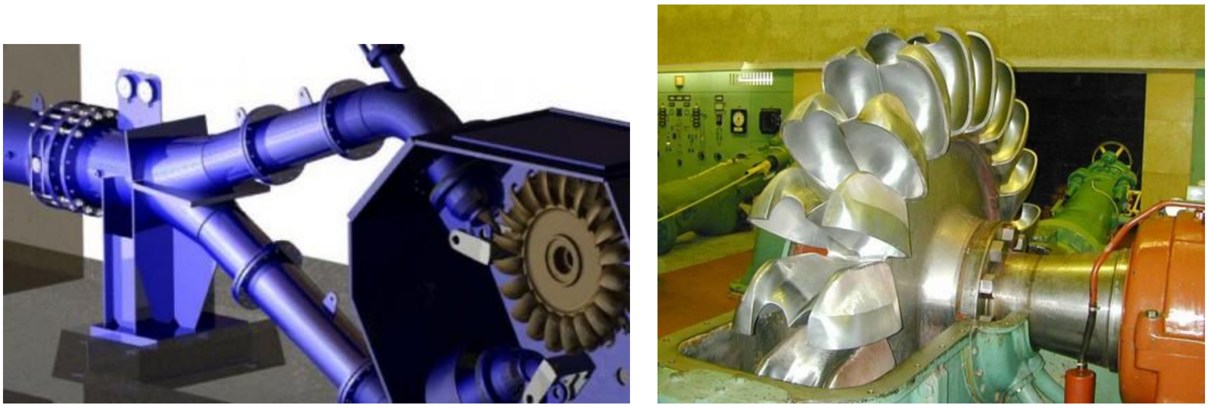
Impulse turbines change the velocity of a water jet. The jet impinges on the turbine's curved blades which change the direction of the flow. The resulting change in momentum causes a force on the turbine blades. Since the turbine is spinning, the force acts through a distance (work) and the diverted water flow is left with diminished energy. Prior to hitting the turbine blades, the water's pressure (potential energy) is converted to kinetic energy by a nozzle and focused on the turbine. No pressure change occurs at the turbine blades, and the turbine doesn't require a housing for operation. Newton's second law describes the transfer of energy for impulse turbines. Impulse turbines are most often used in very high (>300m/984ft) head applications

Impulse turbines: Pelton turbine.

Pelton turbines

A Pelton turbine consists of a set of buckets or cups mounted around a hub. Pelton turbines are not immersed in water. Instead, a Pelton turbine operates in air with the wheel driven by jets of high pressure water hitting the buckets or cups. Kinetic energy of the water jets is transferred to the turbine.

The Pelton wheel is among the most efficient types of water turbines. It was invented by Lester Allan Pelton (1829-1908) in the 1870s, and is an impulse machine, meaning that it uses the principle of Newton's second law to extract energy from a jet of fluid.



Pelton turbine

Although the one-piece cast impulse water turbine was invented by Samuel Knight in Sutter Creek, in the California Mother Lode gold mining region, Pelton modified this invention to create his more efficient design. Knight Foundry is the last water-powered foundry known to exist in the United States and is still operated using Knight Impulse turbines, used to extract power from high heads and low discharge water flows.

The free-jet turbine or Pelton wheel, a type of impulse turbine, named after L. A. Pelton who invented it in 1880. Water passes through nozzles and strikes spoon-shaped buckets or cups arranged on the periphery of a runner, or wheel, which causes the runner to rotate, producing mechanical energy. The runner is fixed on a shaft, and the rotational motion of the turbine is transmitted by the shaft to a generator.

Pelton turbines are suited to high head, low flow applications. Typically, to work this type of turbine, water is piped down a hillside so that at the lower end of the pipe it emerges from a narrow nozzle as a jet with very high velocity. The Pelton turbine can be controlled by adjusting the flow of water to the buckets. In order to stop the wheel a valve is used to shut off the water completely. Small adjustments, necessitated by alterations in the load on the generator, are more safely made by a device which deflects part of the water jet away from the buckets. Pelton wheels are used in storage power stations with downward gradients up to 2,000 meters and can contain up to 6 nozzles.

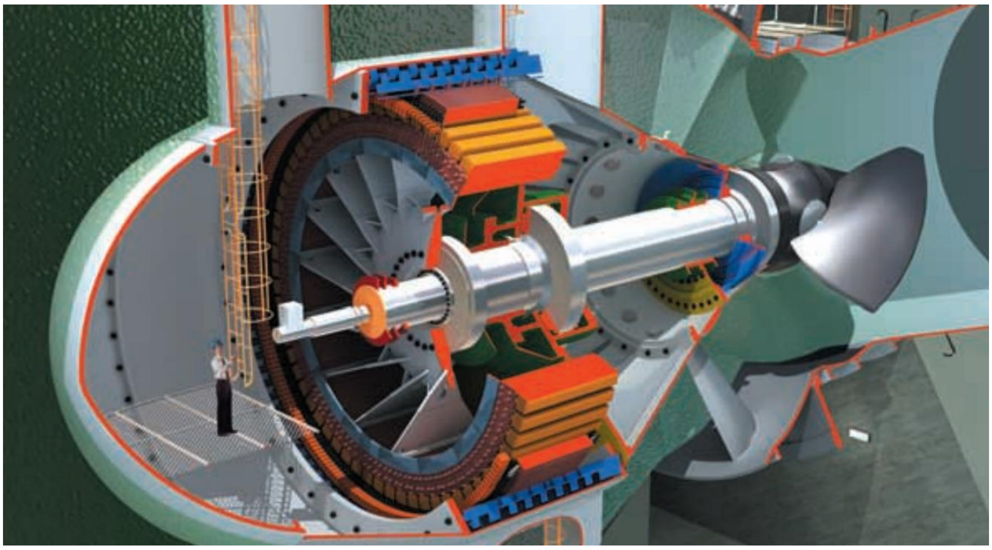
The water flows along the tangent to the path of the runner. Nozzles direct forceful streams of water against a series of spoon-shaped buckets mounted around the edge of a wheel. As water flows into the bucket, the direction of the water velocity changes to follow the contour of the bucket. When the water-jet contacts the bucket, the water exerts pressure on the bucket and the water is decelerated as it does a "U-turn" and flows out the other side of the bucket at low velocity. In the process, the water's momentum is transferred to the turbine. This "impulse" does work on the turbine. For maximum power and efficiency, the turbine system is designed such that the water-jet velocity is twice the velocity of the bucket. A very small percentage of the water's original kinetic energy will still remain in the water; however, this allows the bucket to be emptied at the same rate it is filled, thus allowing the water flow to continue uninterrupted.

Often two buckets are mounted side-by-side, thus splitting the water jet in half. This balances the side-load forces on the wheel, and helps to ensure smooth, efficient momentum transfer of the fluid jet to the turbine wheel. Because water and most liquids are nearly incompressible, almost all of the available energy is extracted in the first stage of the hydraulic turbine. Therefore, Pelton wheels have only one turbine stage, unlike gas turbines that operate with compressible fluid. Pelton wheels are the preferred turbine for hydro-power, when the available water source has relatively high hydraulic head at low flow rates. Pelton wheels are made in all sizes. There exist multi-ton Pelton wheels mounted on vertical oil pad bearings in hydroelectric plants. The largest units can be up to 200 megawatts. The smallest Pelton wheels are only a few inches across, and can be used to tap power from mountain streams having flows of a few gallons per minute. Some of these systems utilize household plumbing fixtures for water delivery. These small units are recommended for use with thirty meters or more of head, in order to generate significant power levels. Depending on water flow and design, Pelton wheels operate best with heads from 15 meters to 1,800 meters, although there is no theoretical limit.

The Pelton wheel is most efficient in high head applications. Thus, more power can be extracted from a water source with high-pressure and low-flow than from a source with low-pressure and high-flow, even though the two flows theoretically contain the same power. Also a comparable amount of pipe material is required for each of the two sources, one requiring a long thin pipe, and the other a short wide pipe.

Bulb turbines

A type of hydro turbine in which the entire generator is mounted inside the water passageway as an integral unit with the turbine. These installations can offer significant reductions in the size of the powerhouse. The bulb turbine is a reaction turbine of Kaplan type which is used for extremely low heads. The characteristic feature of this turbine is that the turbine components as well as the generator are housed inside a bulb, from which the name is developed.



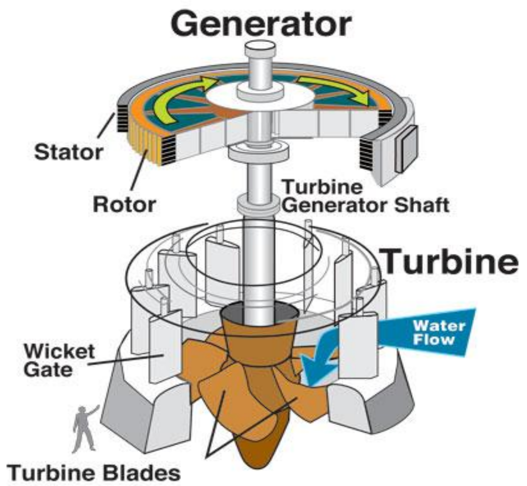
Cross-Sectional view of Bulb Turbine

The main difference from the Kaplan turbine is that the water flows in a mixed axial-radial direction into the guide vane cascade and not through a scroll casing. The guide vane spindles are normally inclined to 60 degree in relation to the turbine shaft and thus results in a conical guide vane cascade contrary to other types of turbines. The bulb turbine is equipped with adjustable wicket gates and adjustable runner blades. This arrangement provides the greatest possible flexibility in adapting to changing net head and changing demands for power output, because the gates and blades can be adjusted to their optimum openings.

The wicket gate operating mechanism is installed inside the inner casing or outside the outer casing; and in both cases, spherical bearings are used to ensure highly smooth operation. The runner blade servomotor is contained in the runner hub, and the pressure oil for it is supplied from the oil head at the upstream end of the generator shaft. While, the runner hub is filled with lubricant oil fed from a gravity tank, which is located on the upper floor of the power house. The runner of a bulb turbine may have different numbers of blades depending on the head and water flow. The bulb turbines have higher full-load efficiency and higher flow capacity as compared to Kaplan turbine. It has a relatively lower construction cost. The bulb turbines can be utilized to tap electrical power from the fast flowing rivers on the hills.

Kaplan turbines

The Kaplan turbine is a propeller-type water turbine that has adjustable blades. It was developed in 1913 by the Austrian professor Viktor Kaplan. The Kaplan turbine was an evolution of the Francis turbine. Its invention allowed efficient power production in low-head applications that was not possible with Francis turbines. Kaplan turbines are now widely used throughout the world in high-flow, low-head power production.



Kaplan turbine and electrical generator cut-away view

Viktor Kaplan living in Brno, Moravia, now Czech Republic, obtained his first patent for an adjustable blade propeller turbine in 1912. But the development of a commercially successful machine would take another decade. Kaplan struggled with cavitation problems, and in 1922 abandoned his research for health reasons. In 1919

Kaplan installed a demonstration unit at Poděbrady, Czechoslovakia. In 1922 Voith introduced an 1100 HP (about 800 kW) Kaplan turbine for use mainly on rivers. In 1924 an 8 MW unit went on line at Lilla Edet, Sweden. This marked the commercial success and wide spread acceptance of Kaplan turbines.

The Kaplan turbine is an inward flow reaction turbine, which means that the working fluid changes pressure as it moves through the turbine and gives up its energy. The design combines radial and axial features. The inlet is a scroll-shaped tube that wraps around the turbine's wicket gate. Water is directed tangentially through the wicket gate and spirals on to a propeller shaped runner, causing it to spin.

The outlet is a specially shaped draft tube that helps decelerate the water and recover kinetic energy. The turbine does not need to be at the lowest point of water flow, as long as the draft tube remains full of water. A higher turbine location, however, increases the suction that is imparted on the turbine blades by the draft tube. The resulting pressure drop may lead to cavitation.

Variable geometry of the wicket gate and turbine blades allow efficient operation for a range of flow conditions. Kaplan turbine efficiencies are typically over 90%, but may be lower in very low head applications. Because the propeller blades are rotated by high-pressure hydraulic oil, a critical element of Kaplan design is to maintain a positive seal to prevent emission of oil into the waterway. Discharge of oil into rivers is not permitted. Current areas of research include CFD driven efficiency improvements and new designs that raise survival rates of fish passing through.

Kaplan turbines are widely used throughout the world for electrical power production. They cover the lowest head hydro sites and are especially suited for high flow conditions. Inexpensive micro turbines are manufactured for individual power production with as little as two feet of head. Large Kaplan turbines are individually designed for each site to operate at the highest possible efficiency, typically over 90%. They are very expensive to design, manufacture and install, but operate for decades.

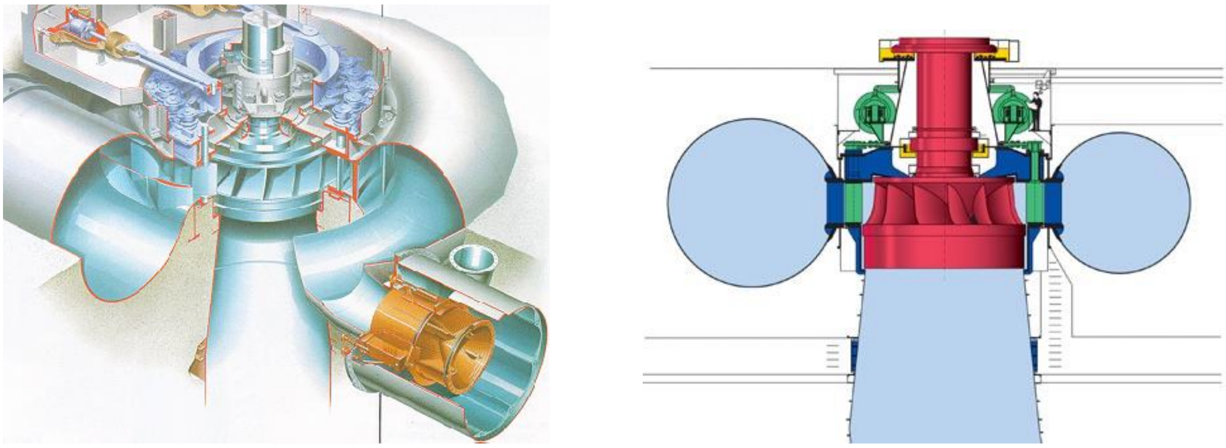
Kaplan turbines have the same challenge as Bulb turbines: both are often installed on large rivers where fish ladders allow the fish to migrate upstream for spawning. Large amounts of fishes have to go through the turbines. Our Kaplan turbines have a “fish friendly” design improving the survival of those species while migrating. Another key environmental concern is accidental water pollution though oil spill. To prevent this, we are experienced with water-lubricated bearings and water filled hubs.

Kaplan turbine generally is the most efficient solution for nominal heads ranging from 20 to 50 meters. Thanks to their double regulation they provide high efficiency over a broad range of head and discharge/output. Compared to bulbs, their vertical configuration allows larger runner diameters (above 10 m.) permitting to increase the unit power and thus minimizing the number of units.

Francis turbines

The Francis turbine is a type of water turbine that was developed by James B. Francis. It is an inward flow reaction turbine that combines radial and axial flow concepts. Francis turbines are the most common water turbine in use today. They operate in a head range of ten meters to several hundred meters and are primarily used for electrical power production.

The Francis turbine is a reaction turbine, which means that the working fluid changes pressure as it moves through the turbine, giving up its energy. A casement is needed to contain the water flow. The turbine is located between the high pressure water source and the low pressure water exit, usually at the base of a dam.



Francis turbine

The inlet is spiral shaped. Guide vanes direct the water tangentially to the turbine wheel, known as a runner. This radial flow acts on the runner's vanes, causing the runner to spin. The guide vanes (or wicket gate) may be adjustable to allow efficient turbine operation for a range of water flow conditions.

As the water moves through the runner its spinning radius decreases, further acting on the runner. For an analogy, imagine swinging a ball on a string around in a circle; if the string is pulled short, the ball spins faster due to the conservation of angular momentum. This property, in addition to the water's pressure, helps Francis and other inward-flow turbines harness water energy efficiently. At the exit, water acts on cup shaped runner features, leaving with no swirl and very little kinetic or potential energy. The turbine's exit tube is shaped to help decelerate the water flow and recover the pressure.

Large Francis turbines are individually designed for each site to operate at the highest possible efficiency, typically over 90%. Francis type units cover a wide head range, from 20 meters to 700 meters, and their output power varies from just a few kilowatts up to one gigawatt. Francis turbines may be designed for a wide range of heads and flows. This, along with their high efficiency, has made them the most widely used turbine in the world.

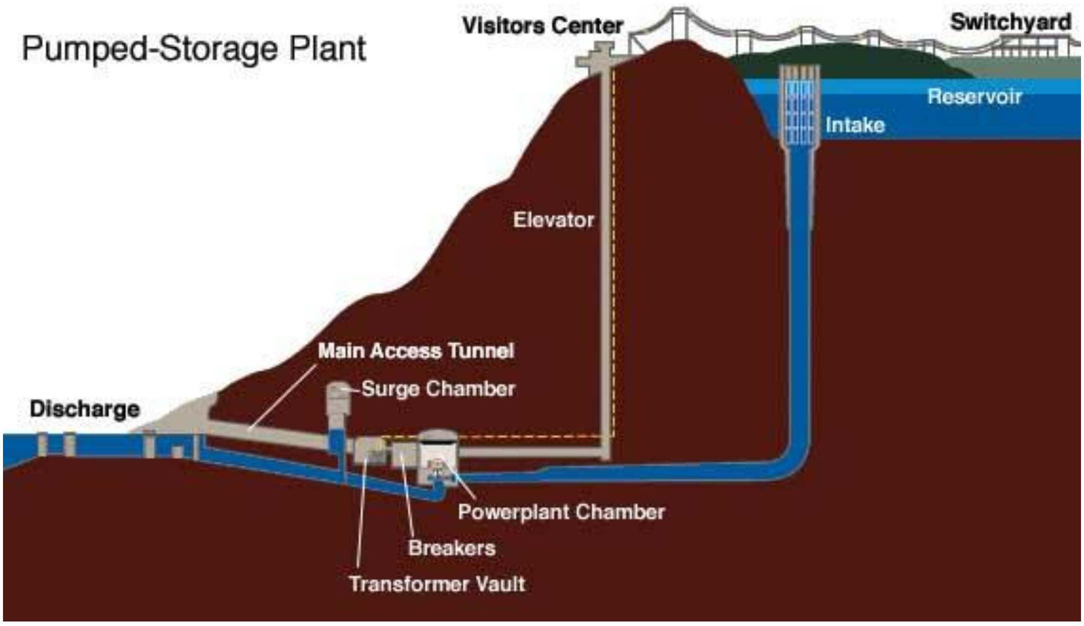
In addition to electrical production, they may also be used for pumped storage; where a reservoir is filled by the turbine (acting as a pump) during low power demand, and then reversed and used to generate power during peak demand.

Francis turbine generally is the most efficient solution for heads ranging from 40 to 600 meters. The runner design is very flexible and can be adapted to get the highest level of efficiency in its whole range of application. Francis Turbines are very robust and able to sustain the high mechanical stress resulting from high heads. Because they are adapted for high head application they often provide the lowest cost per installed MW. With about 60% of the global hydropower capacity in the world, Francis turbines are by far the most widely used type of turbine.

Pump turbines

Very tiny amounts of electrical energy can be stored in capacitors. These are extensively used in electronic circuits, but are impractical to store large quantities of energy. The only alternative is to convert electricity into another form of energy, which can be stored easily. It must also be possible to reconvert the stored energy back into electricity, when it is required. The most widely used method is to store electricity as hydro energy in pumped Hydro Storage schemes. These normally consist of two water reservoirs or lakes at different heights. These are connected by large diameter pipes or tunnels. Reversible pump / turbine machines are located in a power house connected to the pipes.

These are first used to pump water from the lower to the upper reservoir, where it is stored as hydro energy. The pumps are powered by large electric motors, which can also act as generators in the reverse direction. When water is released from the upper reservoir, it flows back down through the reversible machines, which now act as turbines. The turbines are connected to reversible motor / generators, which were initially used as motors to drive the pumps, but now act as generators powered by the turbines and reconvert the hydro energy into electricity.



Pumped storage plant

Pump storage schemes use pumped storage plants. This is the only practical way at present of storing “electricity” on a large scale. The idea is simply to use surplus electricity during off-peak periods to pump water to a mountain top reservoir. This water can also be used as a supplement for other water schemes.

In the event of a shortage of supply of electricity from other power generating stations, the top reservoir can be emptied very quickly back down through the turbine to regenerate electricity. In other words, the motor, which was driving a pump, becomes a generator driven by a turbine. Power generation from these plants is limited as they rely on the water level of the dams or rivers, which in turn is affected by the rainfall in its catchment area. Since pumped storage power stations were constructed as a powerful and economical measure for the storage of electric power, various efforts have been undertaken to improve the economy and the reliability of these power stations. Recent technology concerning single-stage pump-turbines has been explored and developed toward realization of higher head and larger machines.

A Pumped-Storage Power Plant offers the only economic and flexible means of storing large amounts of excess energy and thus allowing the plant owner to efficiently manage the balance between energy production and consumption levels. Like all hydro turbines, Pump turbines can respond almost instantaneously to sharp variations in power demand. This feature is a key contributor to the quality of electricity supplied by a network.