Pump Performance

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Latest revision: 11 January 2012

Nomenclature

bhp	brake horsepower (shaft power delivered to the pump)
C_H	head coefficient: $C_H = gH/\left(\omega^2 D^2\right)$
C_P	power coefficient: $C_P = bhp/\left(\rho\omega^3 D^5\right)$
C_Q	capacity coefficient: $C_Q = \dot{V}/\left(\omega D^3\right)$
D	diameter of impeller blades; for our centrifugal pump, D = 3.5 inches = 0.0889 m
E	Electrical voltage supplied to the DC pump motor
g	gravitational constant (9.81 m/s^2)
H	net head across a pump in terms of the column height of the liquid in the pump
HGL	hydraulic grade line (measured with a piezometer tube)
I	Electrical current supplied to the DC pump motor
$\dot{n}$	rotational speed of the pump shaft (typically in rpm – rotations per minute)
P	static pressure
RPM	shaft rotation rate in units of rotations per minute
T	shaft torque supplied to the pump by the electric motor
V	average velocity in a pipe
V	volume
$\dot{V}$ or Q	volume flow rate
$\dot{W}_{\text{electric}}$	electrical power required to drive the motor that runs a pump, $\dot{W}_{\text{electric}} = EI$
$\dot{W}_{\text{water horsepower}}$	useful power delivered by the pump to the fluid, $\dot{W}_{\text{water horsepower}} = \rho g H \dot{V}$
z	elevation in vertical direction
α	kinetic energy correction factor in energy equation for a control volume
ε	average surface roughness height in a pipe or on the surface of a body
η_{pump}	pump efficiency (efficiency of the pump alone)
$\eta_{\text{pump-motor}}$	pump-motor efficiency (efficiency of the pump *and* motor taken as a unit)
μ	coefficient of dynamic viscosity (also called simply the viscosity)
ν	coefficient of kinematic viscosity (for water, $\nu \approx 1.0 \times 10^{-6}$ m^2/s at room temperature)
ρ	density of the fluid (for water, $\rho \approx 998$ kg/m^3 at room temperature)
ω	angular velocity of the pump shaft (in radians per second)

Educational Objectives

1. Measure the pump performance of a centrifugal pump operating at two rotational speeds.
2. Reinforce knowledge of the usefulness of dimensional analysis – here applied to pump performance curves.

Equipment

1. pump flow rig
2. Himmelstein torque meter and display unit with optical tachometer and Watt meter
3. Validyne pressure transducer with display and analog DC voltage output
4. magnetic resonance flow meter (Rosemount – reads volume flow rate in liters per minute)
5. personal computer with digital data acquisition software

Background

A. Pump Parameters

A centrifugal pump is a turbomachine that adds energy to a fluid by rapid rotation of impeller blades in a snail-shaped housing. Fluid enters the pump through the "eye" of the casing, along the axis of rotation, as shown in Fig. 1.

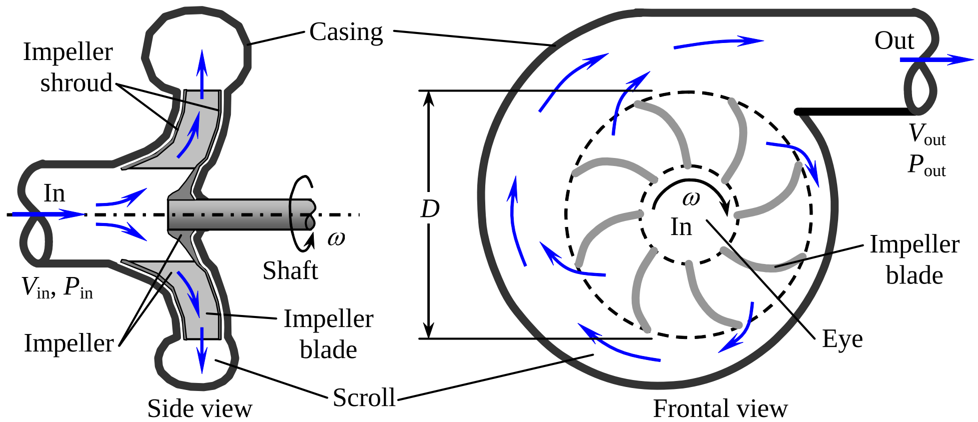

Figure 1. Side view and frontal views of a typical *centrifugal pump*; fluid enters axially in the middle of the pump (the *eye*), is flung around to the outside by the rotating blade assembly (*impeller*), is diffused in the expanding diffuser (*scroll*), and is discharged out the side of the pump. We define *D* as the diameter of the impeller blade.

The spinning impeller blades push the fluid tangentially, causing it to flow radially outward due to so-called "centrifugal forces". (Note that the proper physical explanation is that there is insufficient *centripetal* acceleration to maintain a circular flow, so the fluid flows radially outward instead.) As the fluid passes through the impeller blades, it gains both velocity and pressure. It then exits into the snail-shaped scroll, where the velocity is decreased and the pressure is further increased by the diffuser effect. Often the inlet and outlet of the centrifugal pump are of equal diameter, as in Figure 1. In such a case, and when the fluid is incompressible, the average velocity at the outlet must be *the same* as that at the inlet. We emphasize this point since many people believe (erroneously) that the purpose of a pump is to increase the fluid's velocity. In fact, the velocity of an incompressible fluid is *unchanged* across a pump with equal inlet and outlet diameters. The real purpose of a pump is rather to increase the *pressure* of the fluid.

The increase in fluid energy across a pump is expressed in terms of the increase of the Bernoulli head. (Recall that "head" is the pressure expressed as an equivalent column height of fluid, and head has the dimension of length.) Head is commonly used as a measure of pressure, although pressure is actually head multiplied by ρg. For steady incompressible flow through a pump, the ***net head***, H, of the pump is defined as

$$H = \left(\frac{P}{\rho g} + \frac{V^2}{2g} + z\right)_{\text{outlet}} - \left(\frac{P}{\rho g} + \frac{V^2}{2g} + z\right)_{\text{inlet}} \tag{1}$$

For the simplified case of a pump with identical inlet and outlet diameters, an incompressible fluid, and negligible change in elevation between the inlet and outlet, Equation (1) simplifies to

$$H = \left(\frac{P}{\rho g}\right)_{\text{outlet}} - \left(\frac{P}{\rho g}\right)_{\text{inlet}} \tag{2}$$

When a static pressure tap (just a small hole) is cut into the pipe wall surrounding the inlet (or outlet), the static pressure P at that location can be measured. Since $P / (\rho g)$ is by definition the "head", a piezometer tube attached to the pressure tap measures the local head corresponding to the static pressure at that location. With two piezometer tubes, the net head H supplied by the pump (Eq. 2) can be measured as simply the column height difference between inlet and outlet, as is shown schematically in Figure 2. Note that the ***hydraulic grade line***, HGL, is defined as the height (from some arbitrary reference datum plane) to which a liquid rises in a piezometer tube. Thus, for a pump with equal inlet and outlet diameters that is pumping an incompressible liquid (like water), net head H is also the difference between outlet and inlet HGL, as also indicated in Figure 2.

In this lab experiment, an *electronic differential pressure transducer* replaces the piezometer tubes, and is utilized to measure the net head across the pump. The analog output voltage produced by the transducer is proportional to the net head, and this voltage is acquired by a computerized data acquisition system.

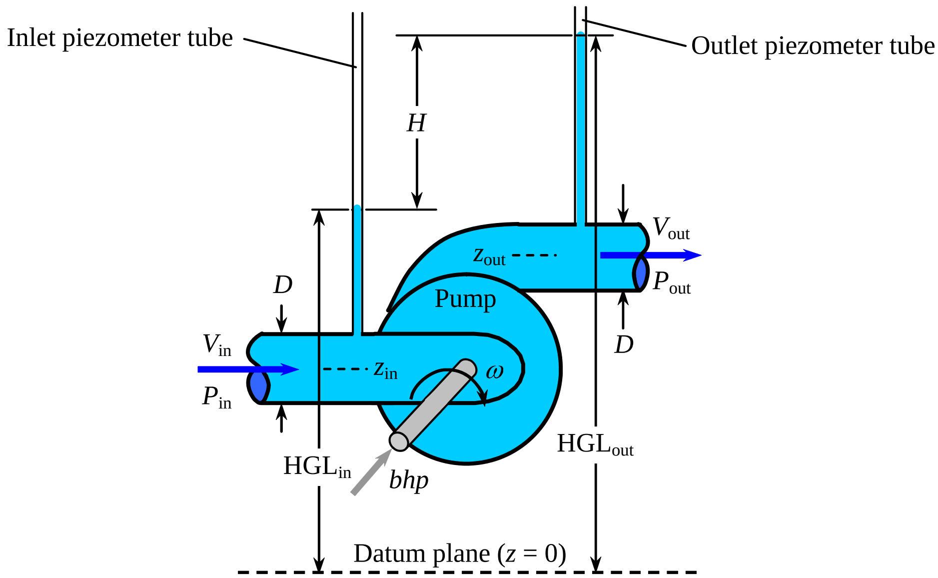

Figure 2. Schematic diagram showing how piezometer tubes are used to measure the net head across a centrifugal pump.

The *useful* power delivered to the fluid by the pump is denoted by $\dot{W}_{\text{water horsepower}}$, the ***water horsepower***,

$$\dot{W}_{\text{water horsepower}} = \rho g H \dot{V} \tag{3}$$

where $\dot{V}$ is the volume flow rate (m^3/s) through the pump. Note that even though we use SI units, $\dot{W}_{\text{water horsepower}}$ is still referred to as "water horsepower". The units of $\dot{W}_{\text{water horsepower}}$ are watts (1 W = 1 N·m/s).

Brake horsepower, bhp, is defined as the actual shaft power required to drive the pump, ωT, where ω is the angular velocity of the shaft (in units of radians per second) and T is the shaft torque (in units of Newton meters). In this lab, both torque, T, and rotational velocity, $\dot{n}$ (in units of rotations per minute, i.e., rpm), are measured at the shaft driving the impeller. After careful conversion of $\dot{n}$ into ω, i.e. $\omega = \dot{n}\cdot(2\pi$ radians/rotation)·(1 minute/60 seconds), brake horsepower is then simply the product of ω and T. Obviously, this power is always *greater* than $\dot{W}_{\text{water horsepower}}$ due to friction and other inefficiencies in the pump. We therefore define η_{pump} as the ***pump efficiency***, i.e., the efficiency of the pump alone,

$$\eta_{\text{pump}} = \dot{W}_{\text{water horsepower}} / \text{bhp} \tag{4}$$

As with all efficiencies, η_{pump} must be less than one. η_{pump} is defined here as the *useful* power actually delivered to the fluid divided by the *required* shaft power to run the pump.

The motor that drives the pump shaft also suffers from inefficiencies. Thus the actual electrical power, $\dot{W}_{\text{electric}}$, required to drive the pump motor is even *larger* than bhp. In the present lab setup, the pump has been separated from the motor so that the brake horsepower and pump efficiencies can be calculated separately from the power required to drive the electric motor. In this lab, we measure both the pump efficiency, η_{pump}, and the ***pump-motor efficiency***, $\eta_{\text{pump-motor}}$, defined as the efficiency of the pump-motor *combination*,

$$\eta_{\text{pump-motor}} = \dot{W}_{\text{water horsepower}} / \dot{W}_{\text{electric}} \tag{5}$$

In other words, the pump-motor efficiency is defined as the useful power delivered to the fluid divided by the required electrical power supplied to the motor. From a pump designer's point of view, the pump efficiency is most important. However, from an economics point of view, it makes sense to discuss the pump-motor efficiency because the user of the pump must *pay* for electrical power; the pump-motor efficiency is thus directly related to the *cost* of operating the pump.

B. Pump Performance Curves

A centrifugal pump is a very complex turbomachine, and purely analytical prediction of its performance is not possible. Hence, performance is obtained empirically, i.e., through experiment. Net head, pump efficiency, and required power are typically plotted as functions of Q or $\dot{V}$, the volume flow rate of fluid through the pump. Plots of this type are called ***pump performance curves***, and typically look like those sketched qualitatively in Figure 3, which is for a centrifugal pump operating at a constant rpm.

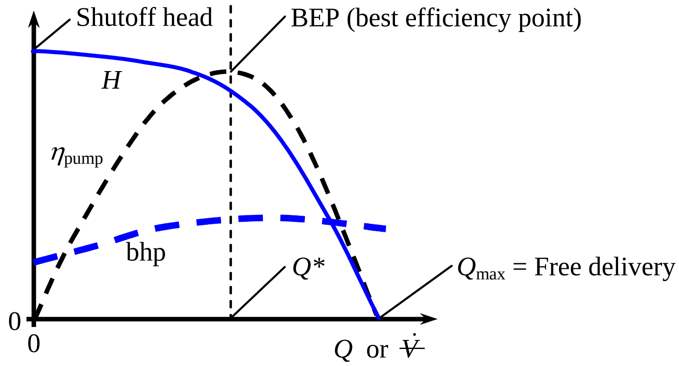

Figure 3. Typical performance curves of a centrifugal pump.

Several interesting observations about Figure 3 can be made. First of all, the net head *H* decreases as volume flow rate increases. In other words, a centrifugal pump can generate a large pressure rise, but only at *low flow rates* (the left side of the plot). When the volume flow rate is zero, the net head is called the ***shutoff head***. Under these conditions, the pump can be unstable, and should not be operated for long periods of time, or the pump motor may burn out. At the other extreme, the maximum flow rate occurs when there is *no* net head (no pressure rise at all!) – this condition is known as the ***free-delivery condition***, and the corresponding volume flow rate is called the ***free delivery***.

A second interesting observation is the shape of the efficiency curve. Combining Equations (3) and (4), we obtain

$$\eta_{\text{pump}} = \frac{\rho g Q H}{\text{bhp}} \quad \text{or} \quad \eta_{\text{pump}} = \frac{\rho g \dot{V} H}{\text{bhp}} \tag{6}$$

It should be obvious, then, why η_{pump} is zero at both extremes of the performance curve. Namely, at zero flow rate, $Q = 0$ and *H* is large, but at $Q = Q_{max}$, $H = 0$. In terms of power, it is useful to remember that work = force · distance, and thus power = work/time = force · velocity. So, when $Q = 0$ and *H* is large, there is a huge force, but no velocity, and hence no useful power. At the other extreme, when $Q = Q_{max}$ but $H = 0$, there is plenty of velocity, but no force (zero net pressure rise), again resulting in no useful power delivered to the fluid. At some intermediate point, η_{pump} reaches a maximum; this point is called the ***best efficiency point***, BEP, and the corresponding flow rate is denoted as *Q**.

Finally, the brake horsepower, bhp, (or alternately the required electrical power to the pump's motor $\dot{W}_{\text{electric}}$) is typically a slowly increasing function of volume flow rate. At zero flow rate, bhp represents power that is all "wasted" since $\eta_{pump} = 0$. In other words, the pump is spinning like mad, churning up the fluid, but not moving any fluid through itself. Under these conditions, bhp, in its entirety, is doing nothing but heating up the fluid and its surroundings. At the other extreme (maximum flow rate), there is no net head, so again the pump is doing no useful work and $\eta_{pump} = 0$, even though bhp is non-zero. Under these conditions, the power supplied to the pump is again wasted in overcoming frictional losses. In fact, when $H = 0$, the pump could theoretically be removed from the system entirely, with no resulting change in flow conditions! Furthermore, it is also possible to "overdrive" the pump; this condition is encountered when the outlet head is *lower* than the inlet head ($H < 0$), and the volume flow rate is greater than Q_{max}. Under these conditions, the pump is being driven by the high volume flow rate, and the supplied electrical power is working *against* the flow. In other words, the pump itself is acting like a *minor loss* in the pipe system; the system would be better off if the pump were not even installed.

C. <u>Similarity Rules</u>

For a given geometrically similar "family" of centrifugal pumps, nondimensionalization of the performance curves is useful, and the technique of *dimensional analysis* is appropriate. In the Precalculations section of this lab write-up, you will perform a dimensional analysis of the pump parameters. The results are

$$\begin{gathered} C_H = C_H(C_Q) \\ C_P = C_P(C_Q) \\ \eta_{\text{pump}} = \eta_{\text{pump}}(C_Q) \quad \text{or} \quad \eta_{\text{pump-motor}} = \eta_{\text{pump-motor}}(C_Q) \end{gathered} \tag{7}$$

where $C_H = \text{Head coefficient} = \dfrac{gH}{\omega^2 D^2}$, $C_Q = \text{Capacity coefficient} = \dfrac{\dot{V}}{\omega D^3}$, and $C_P = \text{Power coefficient} = \dfrac{bhp}{\rho \omega^3 D^5}$.

References

1. Çengel, Y. A. and Cimbala, J. M., *Fluid Mechanics – Fundamentals and Applications*, McGraw-Hill, NY, 2006.
2. White, F. M., *Fluid Mechanics*, Ed. 5, McGraw-Hill, NY, 2003.