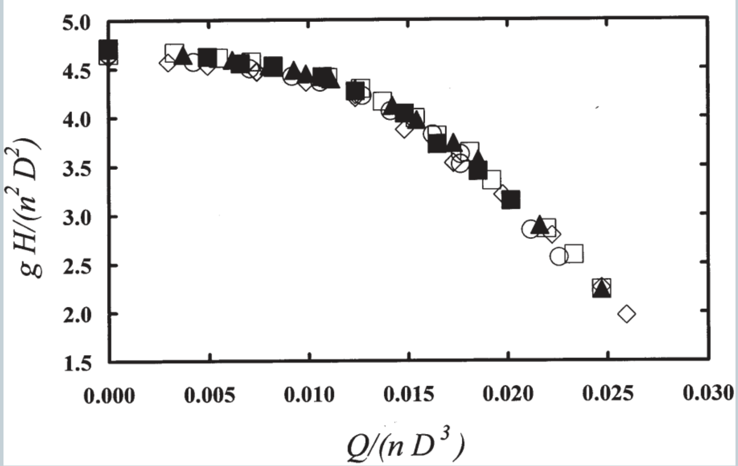
Characteristic Curves

If the speed **N** of a pump or blower with a nominal diameter **D** is constant, the total head **H**, the power **P**, and the efficiency **η**, are exclusive functions of the discharge **Q**.

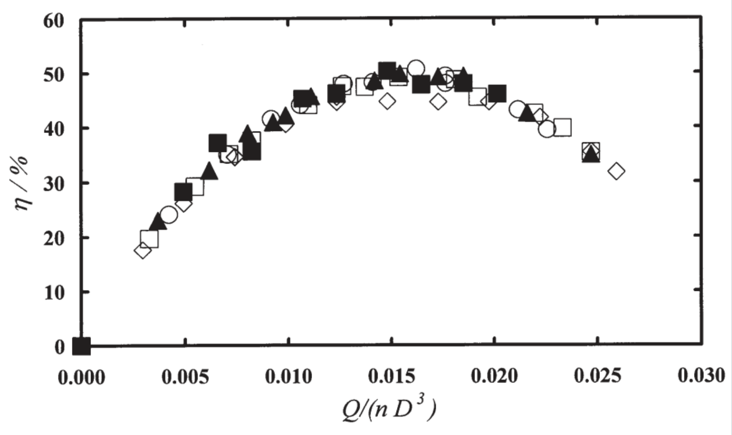
$$H = f_1(Q), \qquad P = f_2(Q), \qquad \eta = f_3(Q)$$

Characteristic Curves

The dimensionless characteristics curves corresponding to the above equations are identical for all geometrically similar pumps and independent of speed.



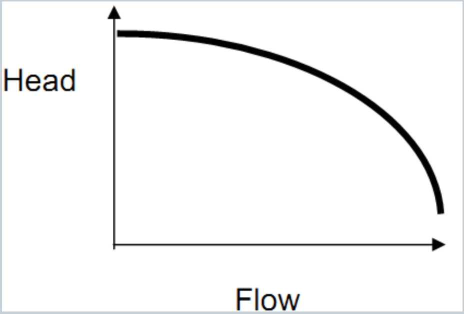
Head coefficient as a function of the flow coefficient for centrifugal pump. ◇ 2000 min^{-1}, Re = 681 800; □ 1800 min^{-1}, Re = 613 600; ▲ 1600 min^{-1}, Re = 545 500; ○ 1400 min^{-1}, Re = 477 300; ■ 1200 min^{-1}, Re =409 100.



Efficiency as a function of the flow coefficient for centrifugal pump. ◇ 2000 min^{-1}, Re = 681 800; □ 1800 min^{-1}, Re = 613 600; ▲ 1600 min^{-1}, Re = 545 500; ○ 1400 min^{-1}, Re = 477 300; ■ 1200 min^{-1}, Re =409 100.

The figure shows a typical head and flow rate curve of a centrifugal pump where the head gradually decreases with increasing flow.

As the resistance of a system increases, the head will also increase. This in turn causes the flow rate to decrease and will eventually reach zero. A zero flow rate is only acceptable for a short period without causing to the pump to burn out.



The theoretical head is

$$H = \frac{u_2 c_{u2}}{g} - \frac{u_1 c_{u1}}{g}$$

for radial entry ($c_{u1}=0$)

$$H = \frac{u_2 c_{u2}}{g}$$

and $c_{u2} = u_2 - \frac{c_{m2}}{\tan\beta_2}$

which, substituted into previous equation, gives

$$H = \frac{u_2}{g}\left(u_2 - \frac{c_{m2}}{\tan\beta_2}\right) = \frac{u_2^2}{g} - \frac{u_2 c_{m2}}{g\tan\beta_2}$$

The meridional velocity c_{m2} is proportional to the capacity Q, equation becomes

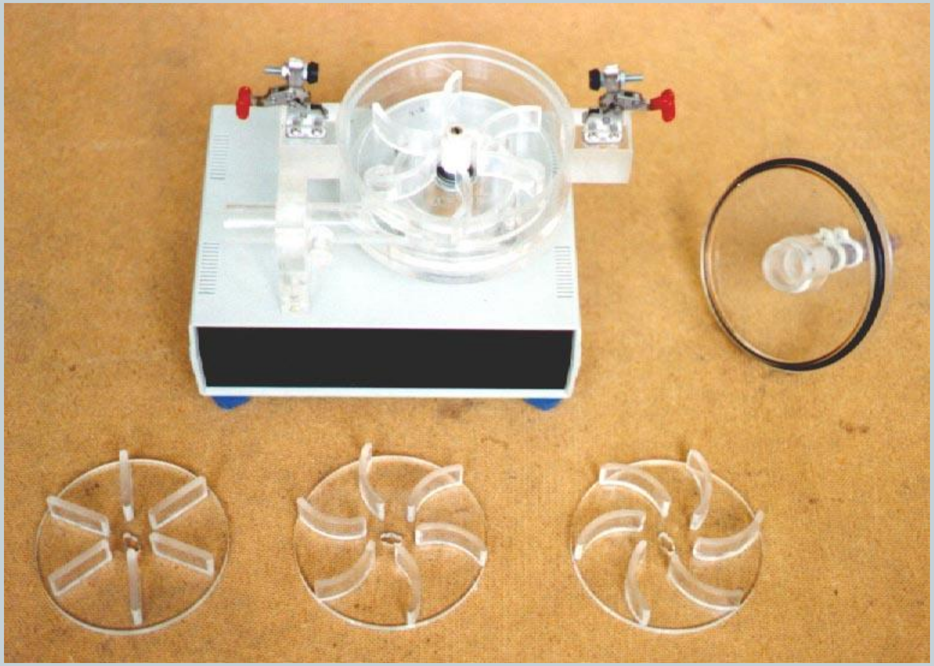
$$H = \frac{u_2^2}{g} - \frac{u_2}{g\tan\beta_2}\frac{Q}{\pi D_2 b_2} = k_1 - k_2 Q$$

In which k_1 and k_2 are constants, with the value of k_2 dependent on the value of the vane angle β_2.

Effect of Blade Outlet Angle

The head-discharge characteristic of a centrifugal pump depends (among other things) on the outlet angle of the impeller blades which in turn depends on blade settings. Three types of blade settings are possible

- the forward facing for which the blade curvature is in the direction of rotation and, therefore, β_2>90°,
- radial, when β_2=90°, and
- backward facing for which the blade curvature is in a direction opposite to that of the impeller rotation and therefore, β_2<90°.



The outlet velocity triangles for all the cases are also shown in below Figure. From the geometry of any triangle, the relationship between c_{u2}, u_2 and β_2 can be written as

$$c_{u2} = u_2 - \frac{c_{m2}}{\tan \beta_2}$$

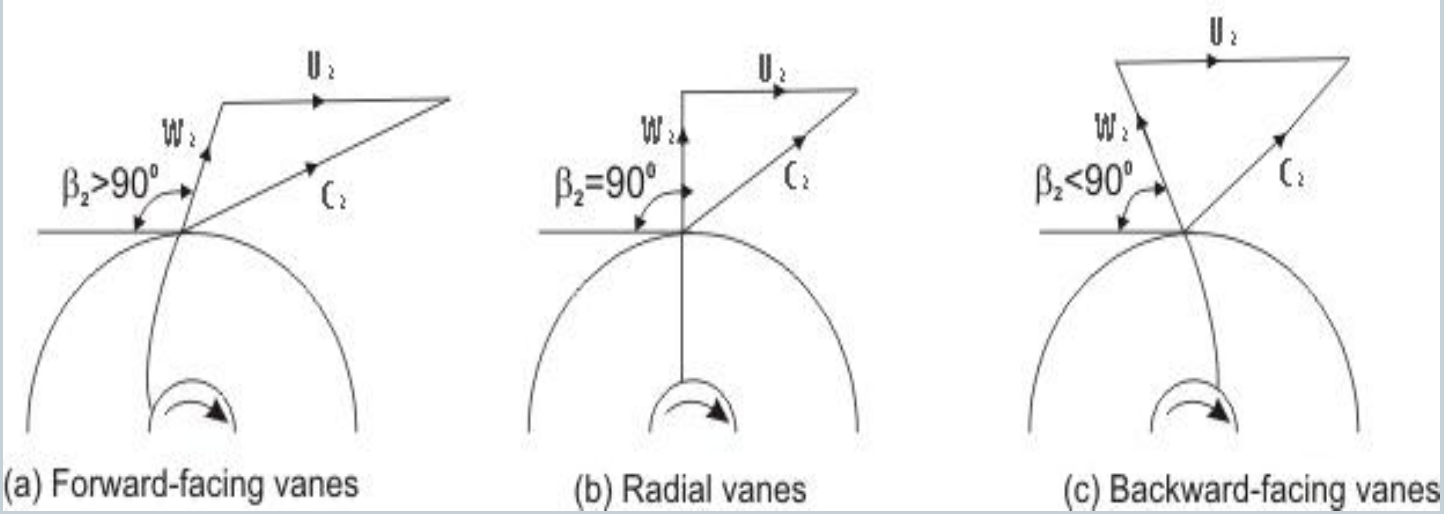


Figure Outlet velocity triangles for different blade settings in a centrifugal pump

$$c_{u2} = u_2 - \frac{c_{m2}}{\tan \beta_2}$$

In case of forward facing blade, $\beta_2 > 90°$ and hence tan β_2 is negative and therefore C_{u2} is more than u_2 . In case of radial blade, $\beta_2 = 90°$ and $C_{u2} = u_2$. In case of backward facing blade, $\beta_2 < 90°$ and $C_{u2} < u_2$, therefore the sign of k_2, the constant in the theoretical head-discharge relationship depends accordingly on the type of blade setting as follows:

For forward curved blades $k_2 < 0$
For radial blades $k_2 = 0$
For backward curved blades $k_2 > 0$

The constant k_1 in the head-discharge relationship is the head at zero flowrate and called **shut-off head**.

$$H_{shut-off} = k_1 = \frac{u_2^2}{g} \; for\ ideal\ case$$

$$H_{shut-off} = \frac{u_2^2}{2g} \; for\ actual\ case$$

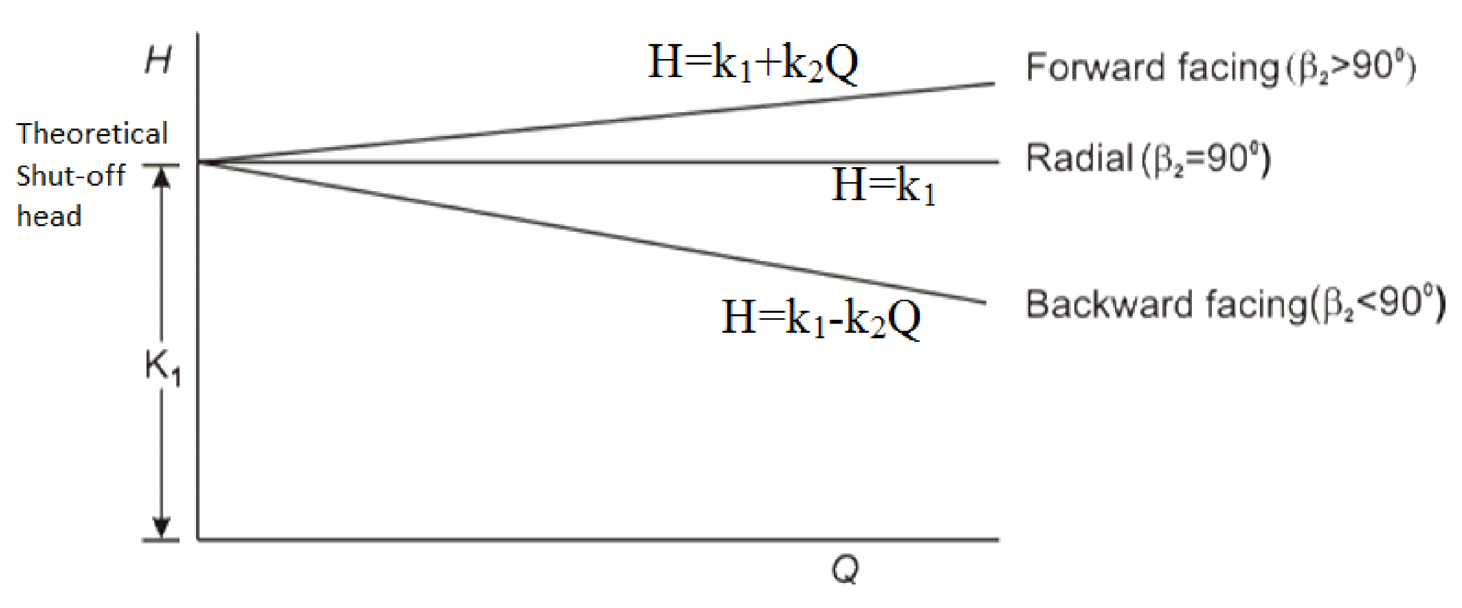


Figure Theoretical head-discharge characteristic curves of a centrifugal pump for different blade settings

The actual characteristics differ from theoretical characteristics due to the certain losses.

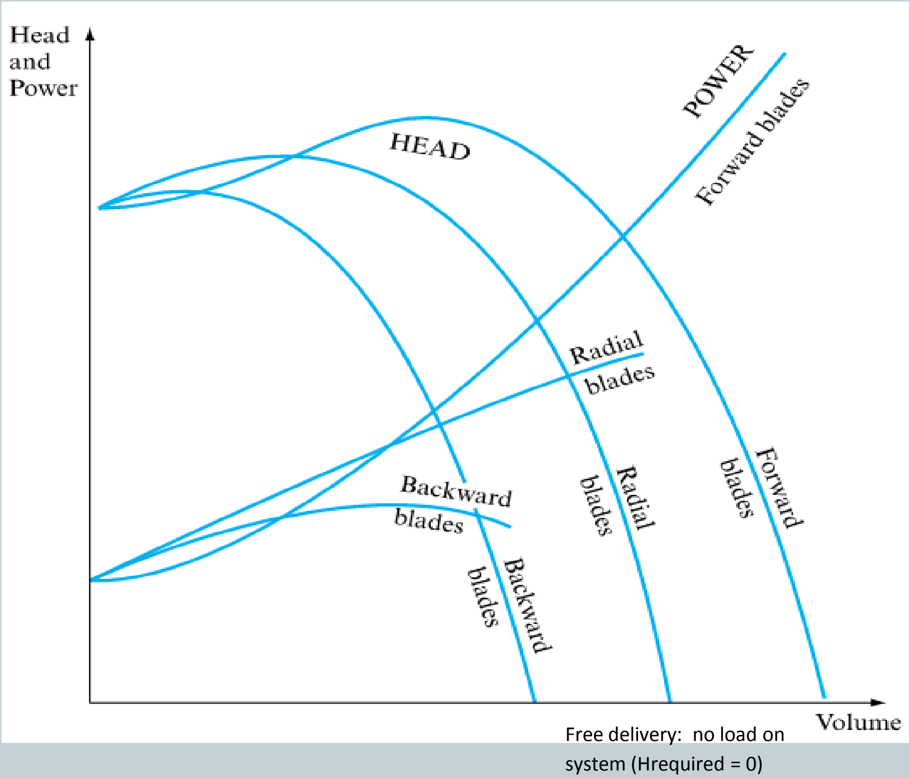
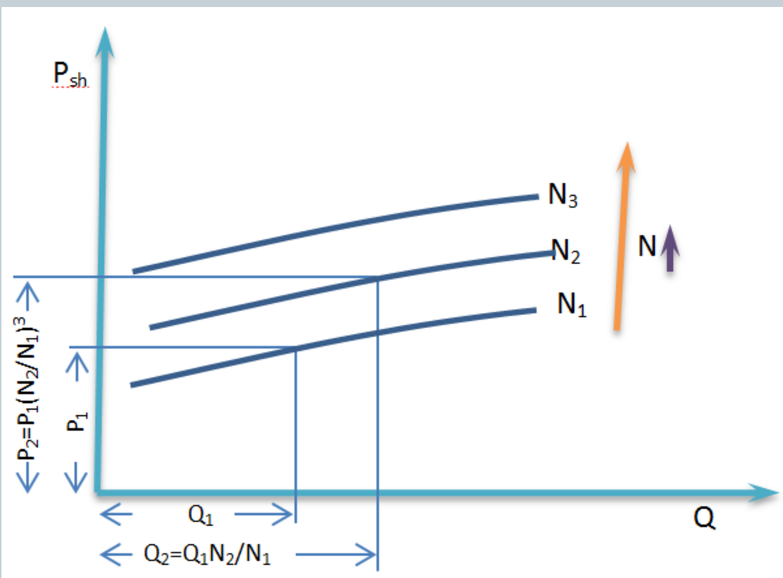


Figure Actual head-discharge and power-discharge characteristic curves of a centrifugal pump

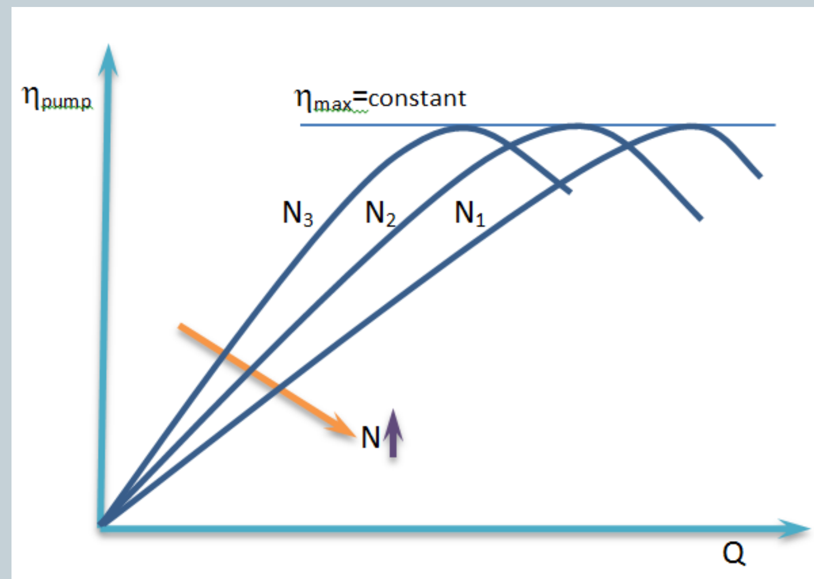
Power required for a pump or blower is (without consideration of losses) is given by

$$P = \gamma QH = \quad PQ = \rho u_2^2 Q - \frac{\rho u_2 Q^2}{\pi D_2 b_2 \tan\beta_2}$$

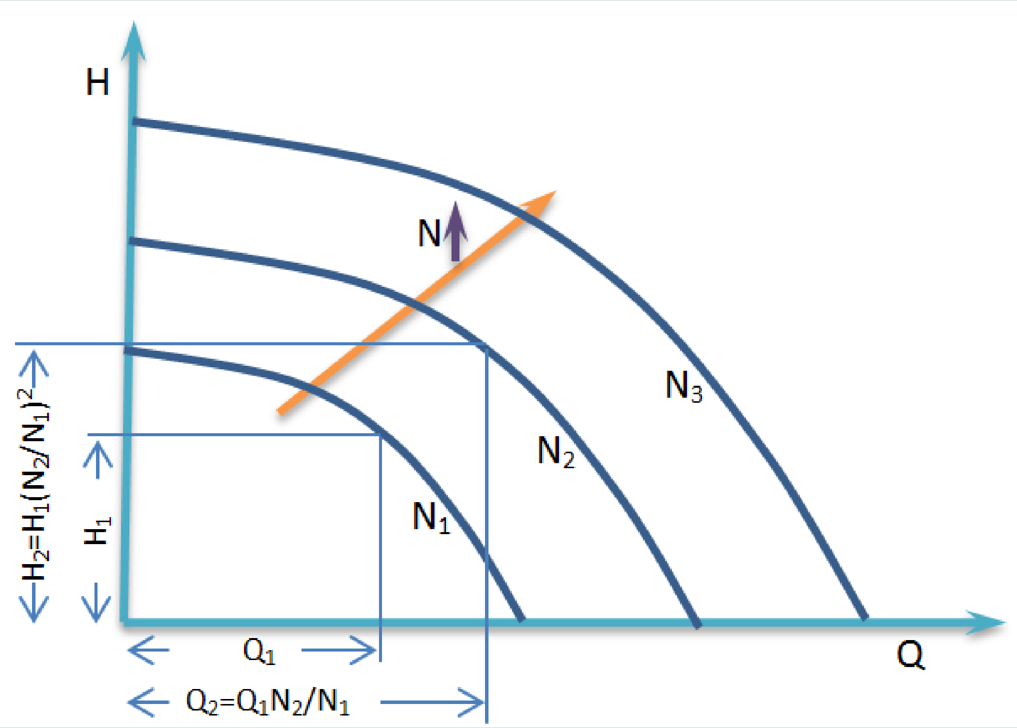
The relationship P=f(Q) is a parabola. Radial and forward curved bladed impellers cause instability during operation for different flow rates there corresponds same head. Since the instabilities, radial and forward curved blades are not preferred. Backward curved blades are preferred in practice because they produce stable operations (performance curves are smooth and for every Q there corresponds a unique H).



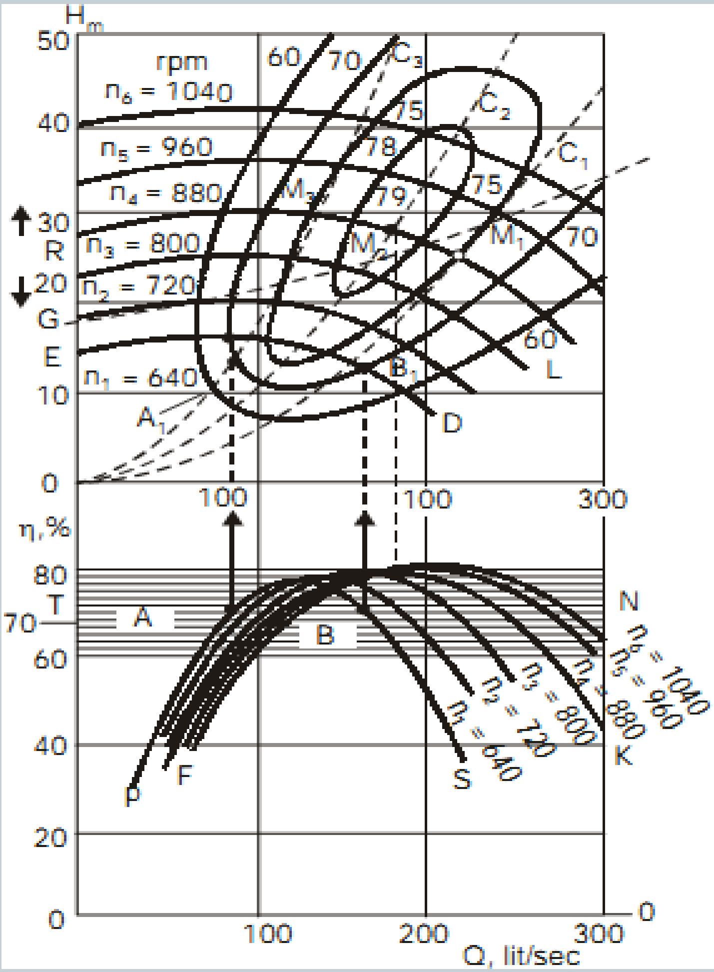
Power-discharge characteristic curves of a centrifugal pump for different speeds



Efficiency-discharge characteristic curves of a centrifugal pump for different speeds



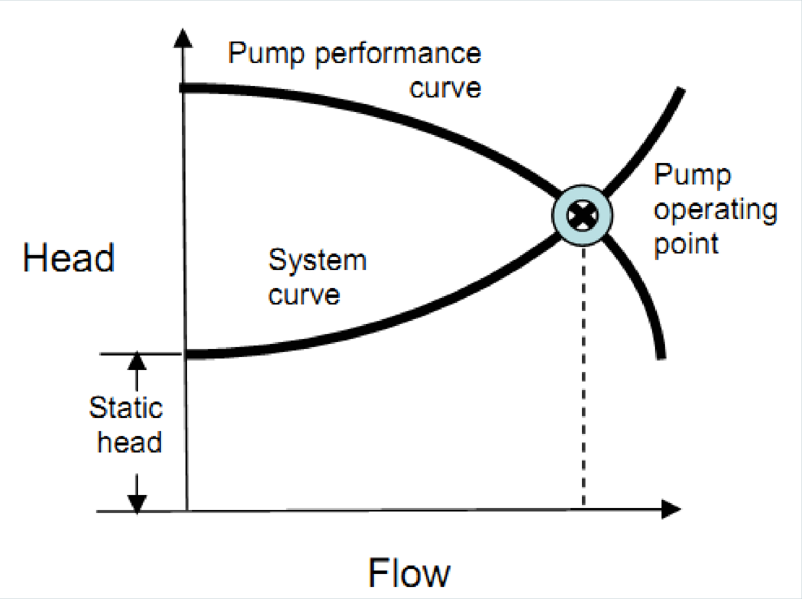
Head-discharge characteristic curves of a centrifugal pump for different speeds



Construction of universal characteristics of pump (H-Q) curve at different speeds and equal efficiencies 'O' curves obtained term (η-Q) curves

System Curve

The rate of flow at a certain head is called the duty point. The pump performance curve is made up of many duty points. The pump operating point is determined by the intersection of the system curve and the pump curve as shown in Figure.

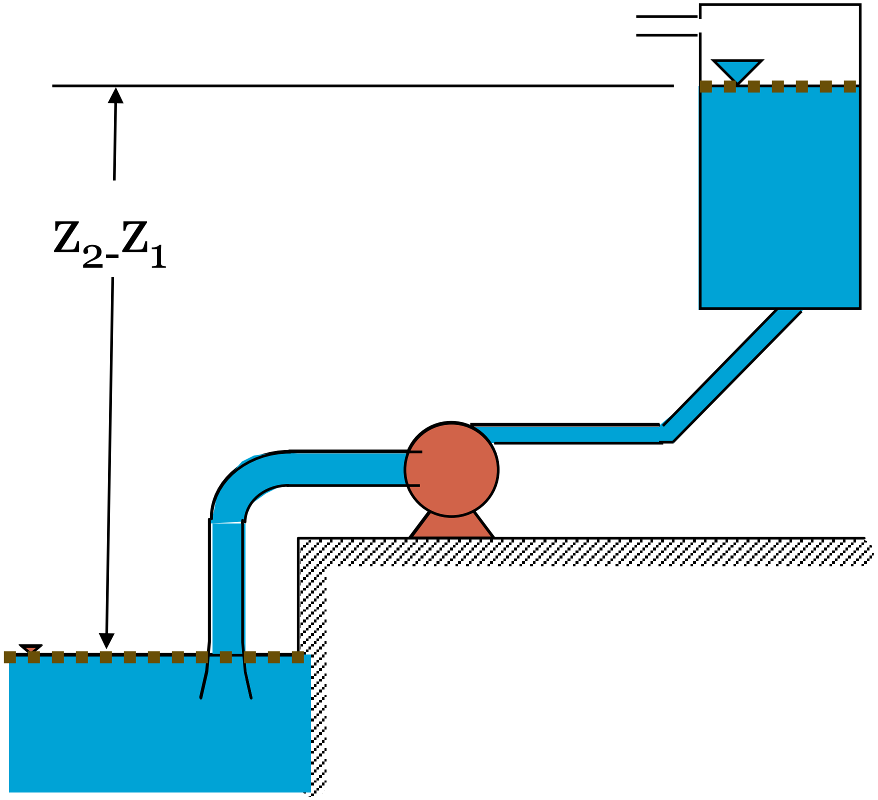


System Curve

$$\frac{p_1}{\gamma} + \frac{V_1^2}{2g} + z_1 + h_p = \frac{p_2}{\gamma} + \frac{V_2^2}{2g} + z_2 + h_l$$

$$h_p = z_2 - z_1 + h_l$$

$$h_l = f\frac{L}{D}\frac{\bar{u}^2}{2g} + (K_1 + K_2 + \ldots + K_n)\frac{\bar{u}^2}{2g}$$



$$h_p = f(Q)$$

often expressed as

$$h_p = a - bQ^2$$

System Curve

Head loss coefficients for a range of pipe fittings

FITTING	LOSS COEFFICIENT K
Gate valve (open to 75 per cent shut)	0.25 → 25
Globe valve	10
Spherical plug valve (fully open)	0.1
Pump foot valve	1.5
Return bend	2.2
90° elbow	0.9
45° elbow	0.4
Large-radius 90° bend	0.6
Tee junction	1.8
Sharp pipe entry	0.5
Radiused pipe entry	→0.0
Sharp pipe exit	0.5

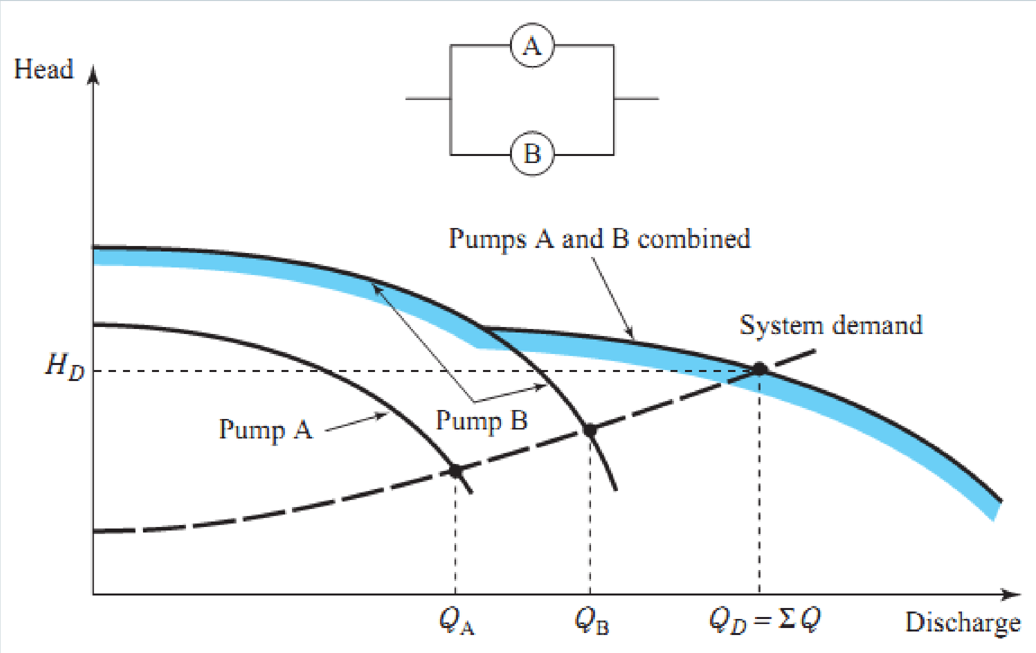
Loss coefficients for sudden contraction

A_2/A_1	0.1	0.3	0.5	0.7	1.0
C_c	0.61	0.632	0.673	0.73	1.0
K	0.41	0.34	0.24	0.14	0

Pumps Operating in Parallel

In some instances, pumping installations may have a wide range of head or discharge requirements, so that a single pump may not meet the required range of demands. In these situations, pumps may be staged either in series or in parallel to provide operation in a more efficient manner. In this discussion, it is assumed that the pumps are placed at a single location with short lines connecting the separate units.

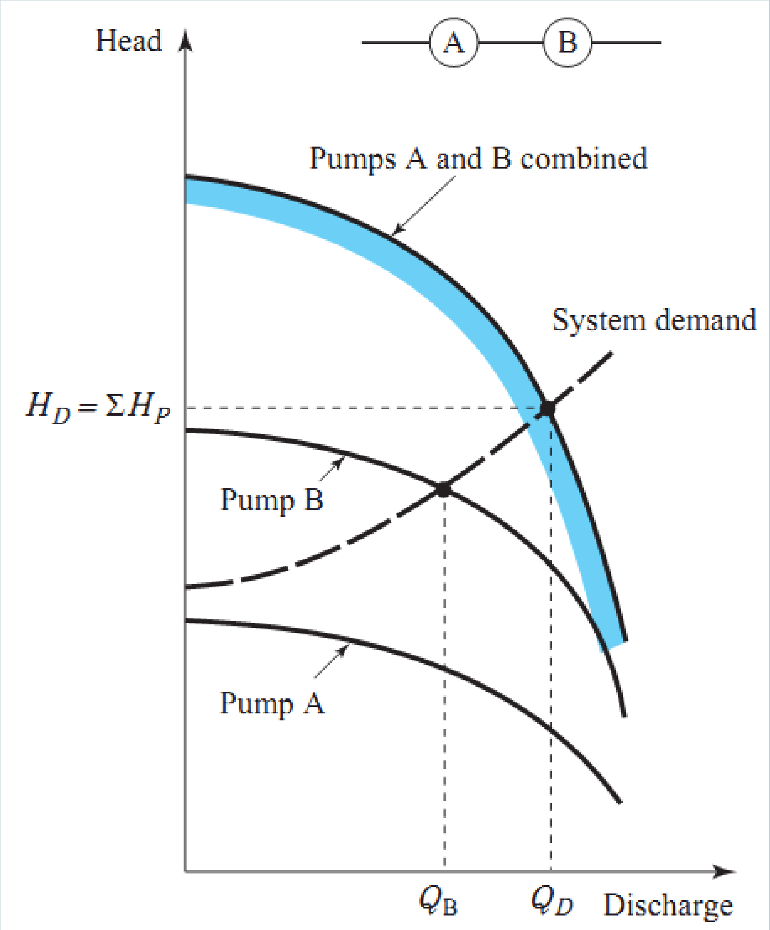
Where a large variation in flow demand is required, two or more pumps are placed in a parallel configuration



Characteristic curves for pumps operating in parallel.

Pumps Operating in Series

For high head demands, pumps placed in series will produce a head rise greater than those of the individual pumps



Characteristic curves for pumps operating in series