Turbomachine Performance

The performance of a pump is usually presented in the form of

- A graph of the head rise, H, as a function of the volume flow rate, Q, for a given speed, Ω or series of speeds. Such a performance graph is known as an "HQ curve".
- Plus a graph of the pump efficiency, η_P, plotted against the volume flow rate, Q.

Typical performance curves for a pump at one speed are sketched in Figure 1. As shown the design operating point is usually identified on these performance curves and commonly coincides with the peak efficiency point.

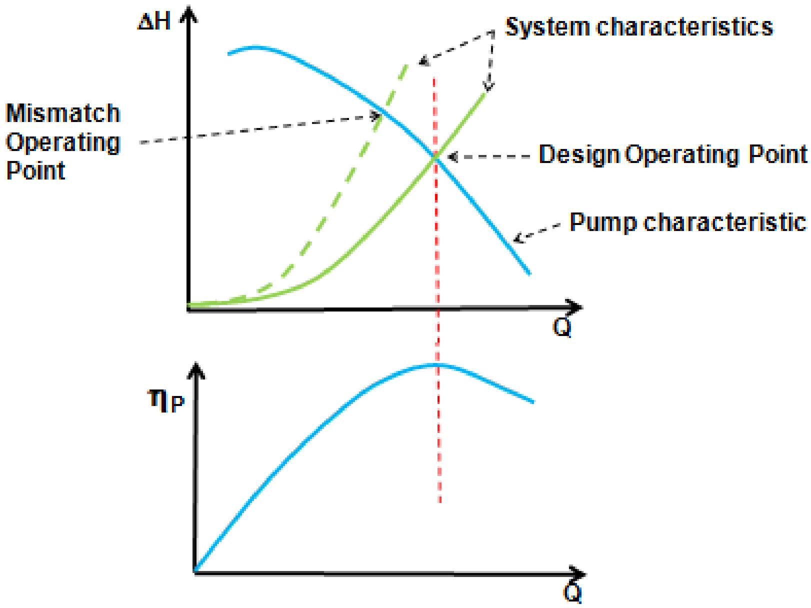

Figure 1: Typical pump performance characteristics.

These performances curves are normally used in the following way during the design or analysis of a hydraulic system. Once the components of the hydraulic system have been identified and their loss coefficients have been assessed, the resistance of the hydraulic system to which the pump is attached or is to be attached can be evaluated. This resistance will be a function of the flow rate, Q, through that system and will take the form of the total head increase, ΔH_s, that would be needed at the proposed location of the pump in order to achieve a particular flow rate, Q, through the hydraulic system. This total head increase will be some function of that flow rate, often proportional to Q^2 since the losses through passive components are roughly proportional to Q^2. This curve, $\Delta H_s(Q)$, when superimposed on the HQ graph of the proposed pump is known as the system characteristic. The condition at which the combination of the pump and the system will operate is given by the intersection of the pump characteristic and the system characteristic. Clearly it is important that this intersection point coincide with the design point (or maximum efficiency point) of the pump. Failure to choose the right pump to ensure that this is the case will not only mean inefficient operation but can also lead to adverse effects such a cavitation, excessive vibration and/or flow instabilities. However, it is not always possible to achieve this when the system demands change with time; then care should be taken to optimize the performance over a range of flows and pump speeds.

In a pump or turbine processing an incompressible fluid, the overall characteristics that are important are the volume flow rate, Q, and the total pressure (or total head) change, $\rho g H$, where H is the total head change. In a pump these dimensional characteristics are conveniently nondimensionalized by defining a head coefficient, ψ,

$$\psi \quad = \quad (p_2^T - p_1^T)/\rho R_{T2}^2 \Omega^2 \quad = \quad gH/R_{T2}^2\Omega^2 \tag{Bfi1}$$

where the impeller tip radius at discharge, R_{T2}, is a common and convenient choice for the characteristic dimension of the machine. Two alternative flow coefficients, ϕ_1 and ϕ_2:

$$\phi_1 = Q/A_1 R_{T1}\Omega \qquad \text{or} \qquad \phi_2 = Q/A_2 R_{T2}\Omega \tag{Bfi2}$$

are also defined where A_1 and A_2 are the inlet and discharge areas, respectively. The discharge flow coefficient is the nondimensional parameter most often used to describe the flow rate. However, in discussions of cavitation, which occurs at the inlet to a pump impeller, the inlet flow coefficient is a more sensible parameter.

Note that with these definitions, (Bfi1) and (Bfi2), the specific speed, N, can be related to the flow and head coefficients by

$$N = \left[\frac{\pi}{\cos\vartheta}\left(1 - \frac{R_{H2}^2}{R_{T2}^2}\right)\right]^{\frac{1}{2}} \frac{\phi_2^{\frac{1}{2}}}{\psi^{\frac{3}{4}}} \tag{Bfi3}$$

where ϑ is the inclination of the impeller discharge flow to the impeller axis and R_{H2} is the impeller hub radius at discharge.