Energy Conversion in Pumps and Turbines

Apart from pipes, most pneumatic or hydraulic systems also involve a whole collection of components such as valves, pumps, turbines, heat exchangers, etc. The flows in these devices are often complicated and frequently require highly specialized analyses. However, effective single phase analyses (homogeneous flow analyses) can also yield useful results and we illustrate this here by reference to work on the multiphase flow through rotating impeller pumps (centrifugal, mixed or axial pumps).

Consistent with the usual turbomachinery conventions, the total pressure increase (or decrease) across a pump (or turbine) and the total volumetric flux (based on the discharge area, A_d) are denoted by Δp^T and j, respectively, and these quantities are non-dimensionalized to form the head and flow coefficients, ψ and ϕ, for the machine:

$$\psi = \frac{\Delta p^T}{\rho \Omega^2 r_d^2} \quad ; \quad \phi = \frac{j}{\Omega r_d} \tag{Nkh1}$$

where Ω and r_d are the rotating speed (in radians/second) and the radius of the impeller discharge respectively and ρ is the mixture density. We note that sometimes in presenting cavitation performance, the impeller inlet area, A_i, is used rather than A_d in defining j, and this leads to a modified flow coefficient based on that inlet area. (A word of caution here; since most pump engineers use total head rather than total pressure it is most important in analyzing the pump performance in multiphase flow to measure the head in height of fluid with the density ρ. Sometimes this is overlooked and the head in height of the suspending fluid is inappropriately used in the calculation of ψ.)

The typical centrifugal pump performance with multiphase mixtures is exemplified by figures 1, 2 and 3. Figure 1 from Herbich (1975) presents the performance of a centrifugal dredge pump ingesting silt/clay/water mixtures with mixture densities, ρ, up to $1380 kg/m^3$. The corresponding solids fractions therefore range up to about 25% and the figure indicates that, provided ψ is defined using the mixture density, there is little change in the performance even up to such high solids fractions. Herbich also shows that the silt and clay suspensions cause little change in the equivalent homogeneous cavitation performance of the pump.

Data on the same centrifugal pump with air/water mixtures of different volume quality, β, is included in figure 2 (Herbich 1975). Again, there is little difference between the multiphase flow performance

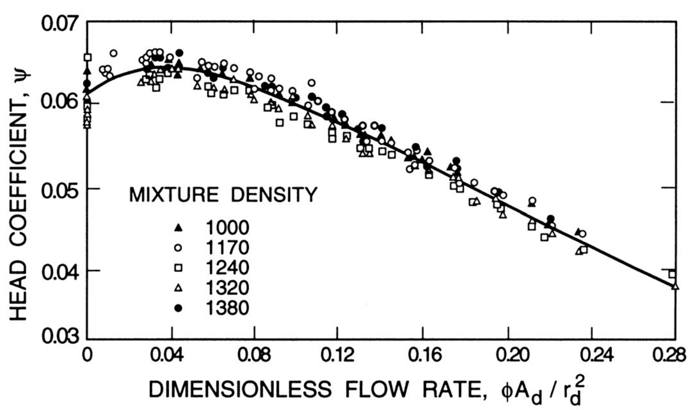

Figure 1: The head coefficient, ψ, for a centrifugal dredge pump ingesting silt/clay/water mixtures plotted against a non-dimensional flow rate, $\phi A_d/r_d^2$, for various mixture densities (in kg/m^3). Adapted from Herbich (1975).

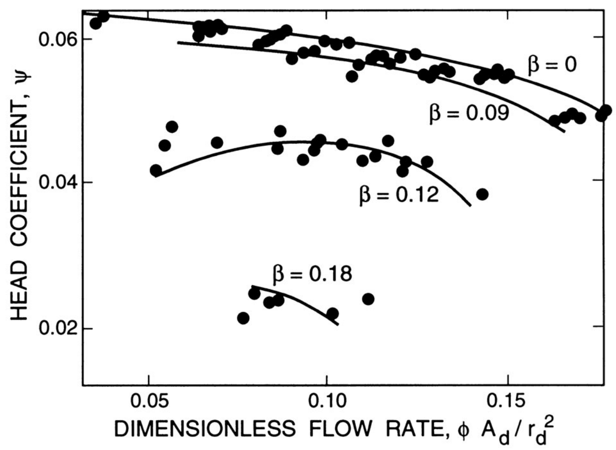

Figure 2: The head coefficient, ψ, for a centrifugal dredge pump ingesting air/water mixtures plotted against a non-dimensional flow rate, $\phi A_d/r_d^2$, for various volumetric qualities, β. Adapted from Herbich (1975).

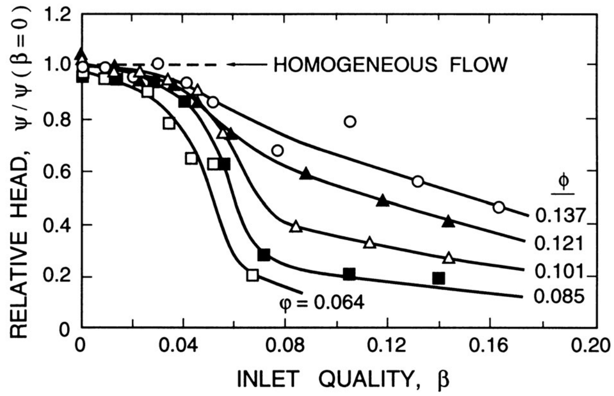

Figure 3: The ratio of the pump head with air/water mixtures to the head with water alone, $\psi/\psi(\beta=0)$, as a function of the inlet volumetric quality, β, for various flow coefficients, ϕ. Data from Patel and Runstadler (1978) for a centrifugal pump.

and the homogeneous flow prediction at small discharge qualities. However, unlike the solids/liquid case, the air/water performance begins to decline precipitously above some critical volume fraction of gas, in this case a volume fraction consistent with a discharge quality of about 9%. Below this critical value, the homogeneous theory works well; larger volumetric qualities of air produce substantial degradation in performance.

Patel and Runstadler (1978), Murakami and Minemura (1978) and many others present similar data for pumps ingesting air/water and steam/water mixtures. Figure 3 presents another example of the air/water flow through a centrifugal pump. In this case the critical inlet volumetric quality is only about $\beta = 3\%$ or 4% and the degradation appears to occur at lower volume fractions for lower flow coefficients. Murakami and Minemura (1978) obtained similar data for both axial and centrifugal pumps, though the performance of axial flow pumps appear to fall off at even lower air contents.

A qualitatively similar, precipitous decline in performance occurs in single phase liquid pumping when cavitation at the inlet to the pump becomes sufficiently extensive. This performance degradation is normally presented dimensionlessly by plotting the head coefficient, ψ, at a given, fixed flow coefficient against a dimensionless inlet pressure, namely the cavitation number, σ (see section **??**), defined as

$$\sigma = \frac{(p_i - p_V)}{\frac{1}{2}\rho_L \Omega^2 r_i^2} \tag{Nkh2}$$

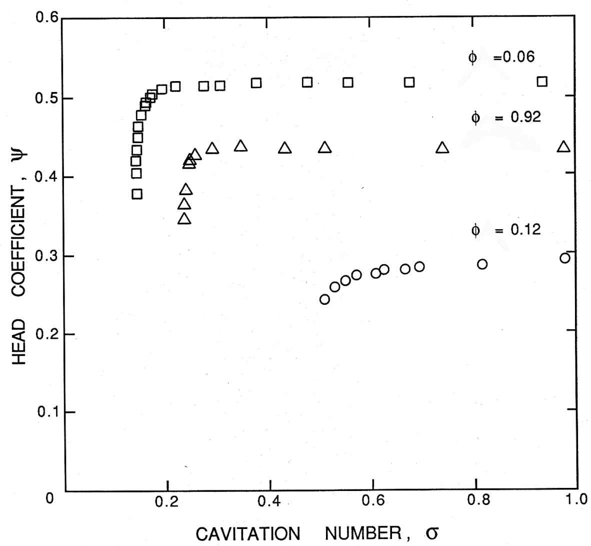

Figure 4: Cavitation performance for a typical centrifugal pump (Franz *et al.* 1990) for three different flow coefficients, ϕ

where p_i and r_i are the inlet pressure and impeller tip radius and p_V is the vapor pressure. An example is shown in figure 4 which presents the cavitation performance of a typical centrifugal pump. Note that the performance declines rapidly below a critical cavitation number that usually corresponds to a fairly high vapor volume fraction at the pump inlet.

There appear to be two possible explanations for the decline in performance in gas/liquid flows above a critical volume fraction. The first possible cause, propounded by Murakami and Minemura (1977,1978), Patel and Runstadler (1978), Furuya (1985) and others, is that, when the void fraction exceeds some critical value the flow in the blade passages of the pump becomes stratified because of the large crossflow pressure gradients. This allows a substantial deviation angle to develop at the pump discharge and, as in conventional single phase turbomachinery analyses (Brennen 1994), an increasing deviation angle implies a decline in performance. The lower critical volume fractions at lower flow coefficients would be consistent with this explanation since the pertinent pressure gradients will increase as the loading on the blades increases. Previously, in section (Njh), we discussed the data on the bubble size in the blade passages compiled by Murakami and Minemura (1977, 1978). Bubble size is critical to the process of stratification since larger bubbles have larger relative velocities and will therefore lead more readily to stratification. But the size of bubbles in the blade passages of a pump is usually determined by the high shear rates to which the inlet flow is subjected and therefore the phenomenon has two key processes, namely shear at inlet that determines bubble size and segregation in the blade passages that governs performance.

The second explanation (and the one most often put forward to explain cavitation performance degradation) is based on the observation that the vapor (or gas) bubbles grow substantially as they enter the pump and subsequently collapse as they are convected into regions of higher pressure within the blade passages of the pump. The displacement of liquid by this volume growth and collapse introduces an additional flow area restriction into the flow, an additional inlet *nozzle* caused by the cavitation. Stripling and Acosta (1962) and others have suggested that the head degradation due to cavitation could be due to a lack of pressure recovery in this effective additional nozzle.