Pump Performance

The performance of a pump when presented nondimensionally will take the generic form sketched in figure

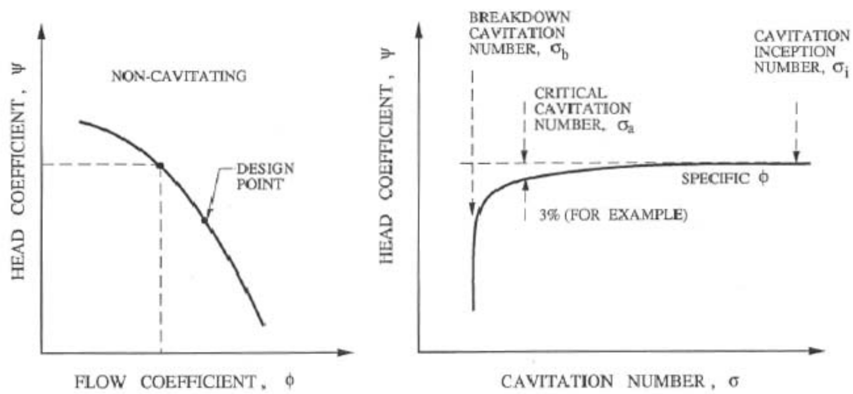

Figure 1: Schematic of noncavitating performance, $\psi(\phi)$, and cavitating performance, $\psi(\phi, \sigma)$, showing the three key cavitation numbers.

1. As discussed earlier, the noncavitating performance will consist of the head coefficient, ψ, as a function of the flow coefficient, ϕ, where the design conditions can be identified as a particular point on the $\psi(\phi)$ curve. The noncavitating characteristic should be independent of the speed, Ω, though at lower speeds there may be some deviation due to viscous or Reynolds number effects. The *cavitating performance*, as illustrated on the right in figure 1, is presented as a family of curves, $\psi(\phi, \sigma)$, each for a specific flow coefficient, in a graph of the head coefficient against cavitation number, σ. Frequently, of course, both performance curves are presented dimensionally; then, for example, the NPSH is often used instead of the cavitation number as the abscissa for the cavitation performance graph.

It is valuable to identify three special cavitation numbers in the cavitation performance graph. Consider a pump operating at a particular flow rate or flow coefficient, while the inlet pressure, $NPSH$, or cavitation number is gradually reduced. As discussed in the previous chapter, the first critical cavitation number to be reached is that at which cavitation first appears; this is called the cavitation inception number, σ_i. Often the occurrence of cavitation is detected by the typical crackling sound that it makes (see section (Mbei)). As the pressure is further reduced, the extent (and noise) of cavitation will increase. However, it typically requires a further, substantial decrease in σ before any degradation in performance is encountered. When this occurs, the cavitation number at which it happens is often defined by a certain percentage loss in the head rise, H, or head coefficient, ψ, as shown in figure 1. Typically a critical cavitation number, σ_a, is defined at which the head loss is 2, 3 or 5%. Further reduction in the cavitation number will lead to major deterioration in the performance; the cavitation number at which this occurs is termed the breakdown cavitation number, and is denoted by σ_b.

It is important to emphasize that these three cavitation numbers may take quite different values, and to confuse them may lead to serious misunderstanding. For example, the cavitation inception number, σ_i, can be an order of magnitude larger than σ_a or σ_b. There exists, of course, a corresponding set of critical suction specific speeds that we denote by S_i, S_a, and S_b. Some typical values of these parameters are presented in table 1 which has been adapted from McNulty and Pearsall (1979). Note the large differences between S_i and S_b.

Perhaps the most common misunderstanding concerns the recommendation of the Hydraulic Institute that is reproduced in figure 2. This suggests that a pump should be operated with a Thoma cavitation

Table 1: Inception and breakdown suction specific speeds for some typical pumps (from McNulty and Pearsall 1979).

PUMP TYPE	N_D	Q/Q_D	S_i	S_b	S_b/S_i
Process pump with	0.31	0.24	0.25	2.0	8.0
volute and diffuser		1.20	0.8	2.5	3.14
Double entry pump	0.96	1.00	<0.6	2.1	>3.64
with volute		1.20	0.8	2.1	2.67
Centrifugal pump w.	0.55	0.75	0.6	2.41	4.02
diffuser and volute		1.00	0.8	2.67	3.34
Cooling water pump	1.35	0.50	0.65	3.40	5.24
(1/5 scale model)		0.75	0.60	3.69	6.16
		1.00	0.83	3.38	4.07
Cooling water pump	1.35	0.50	0.55	2.63	4.76
(1/8 scale model)		0.75	0.78	3.44	4.40
		1.00	0.99	4.09	4.12
		1.25	1.07	2.45	2.28
Cooling water pump	1.35	0.50	0.88	3.81	4.35
(1/12 scale model)		0.75	0.99	4.66	4.71
		1.00	0.75	3.25	4.30
		1.25	0.72	1.60	2.22
Volute pump	1.00	0.60	0.76	1.74	2.28
		1.00	0.83	2.48	2.99
		1.20	1.21	2.47	2.28

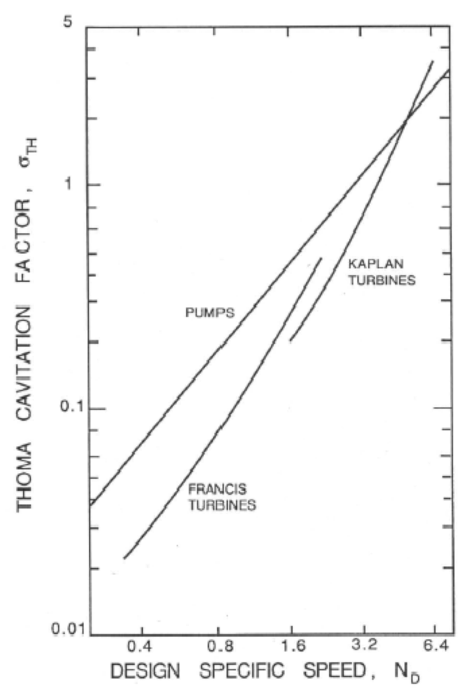

Figure 2: The Hydraulic Institute standards for the operation of pumps and turbines (Hydraulic Institute 1965).

factor, σ_{TH}, in excess of the value given in the figure for the particular specific speed of the application. The line, in fact, corresponds to a critical suction specific speed of 3.0. Frequently, this is erroneously

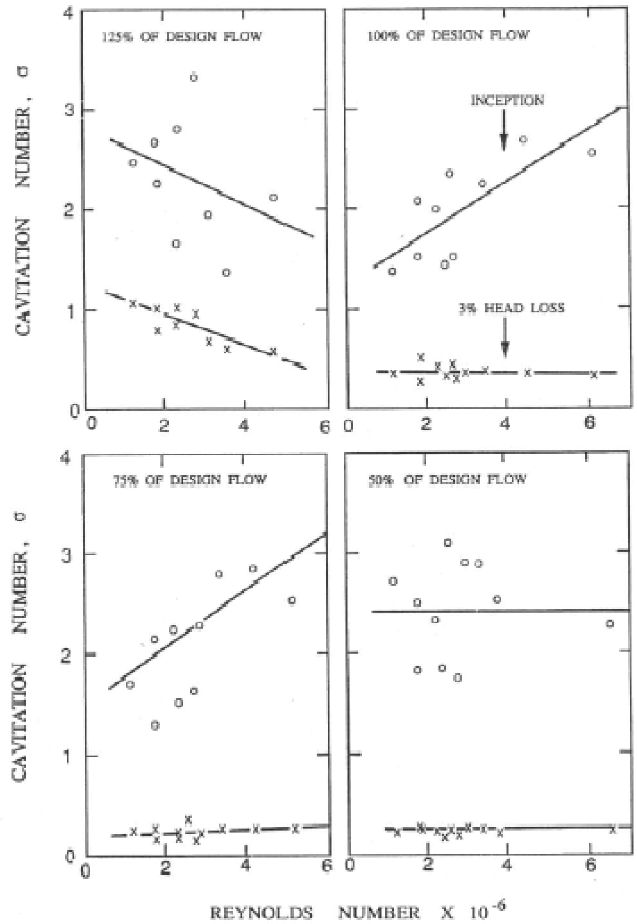

Figure 3: Inception and 3% head loss cavitation numbers plotted against a Reynolds number (based on w_{T1} and blade chord length) for four flow rates (from McNulty and Pearsall 1979).

interpreted as the value of S_i. In fact, it is more like S_a; operation above the line in figure 2 does *not* imply the absence of cavitation, or of cavitation damage.

Data from McNulty and Pearsall (1979) for σ_i and σ_a in a typical pump is presented graphically in figure 3 as a function of the fraction of design flow and the Reynolds number (or velocity). Note the wide scatter in the inception data, and that no clear trend with Reynolds number seems to be present.

The next section will include a qualitative description of the various forms of cavitation that can occur in a pump. Following that, the detailed development of cavitation in a pump will be described, beginning in section (Mbeg) with a discussion of inception.