Turbomachine Design

At the beginning of any turbomachine design process, neither the size nor the shape of the machine is known. The task the turbomachine is required to perform is to employ a shaft rotating at a frequency, Ω (in rad/s), to accept or produce a certain flow rate, Q (in m^3/s) with an associated a head rise or drop, H (in m). As in all fluid mechanical formulations, one should first seek a nondimensional parameter (or parameters) which distinguishes the nature of this task. In this case, there is one and only one nondimensional parametric group that is appropriate and this is known as the **specific speed**, denoted by N. The form of the specific speed is readily determined by dimensional analysis:

$$N = \frac{\Omega Q^{\frac{1}{2}}}{(gH)^{\frac{3}{4}}} \qquad \text{(Bfh1)}$$

Though originally constructed to allow evaluation of the shaft speed associated with a particular head and flow, the name **specific speed** is slightly misleading, because N is just as much a function of flow rate and head rise or drop as it is of shaft speed. Perhaps a more general name, like **the basic performance parameter**, would be more appropriate. Note that the specific speed is a size-independent parameter, since the size of the machine is not known at the beginning of the design process.

In the above definition of the specific speed we have employed a consistent set of units, so that N is

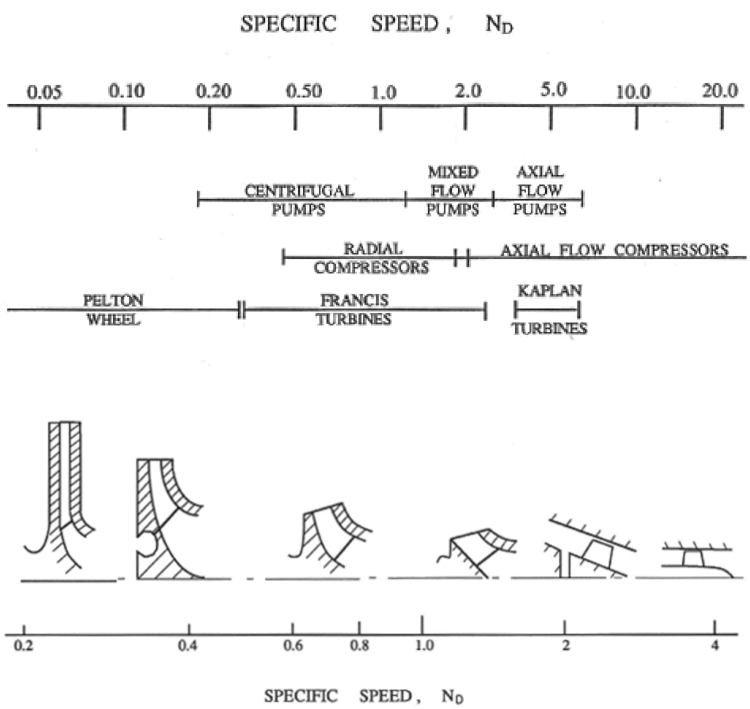

Figure 1: Ranges of specific speeds for typical turbomachines and typical pump geometries for different design speeds (from Sabersky, Acosta and Hauptmann 1989).

truly dimensionless. With these consistent units, the values of N for most common turbomachines lie in the range between 0.1 and 4.0 (see below). Unfortunately, it has been traditional in industry to use an inconsistent set of units in calculating N. In the USA, the g is dropped from the denominator, and values for the speed, flow rate, and head in rpm, gpm, and ft are used in calculating N. This yields values that are a factor of 2734.6 larger than the values of N obtained using consistent units. The situation is even more confused since the Europeans use another set of inconsistent units (rpm, m^3/s, head in m, and no g) while the British employ a definition similar to the U.S., but with Imperial gallons rather than U.S. gallons. One can only hope that the pump and turbine industries would cease the use of these inconsistent measures that would be regarded with derision by any engineer outside of the industry. In these pages, we shall use the dimensionally consistent and, therefore, universal definition of N.

Since turbomachines are designed for specific tasks, the subscripted N_D will be used to denote the design value of the specific speed for a given machine and the design process begins with this value. It is therefore not surprising that the overall or global geometries that have evolved over many decades, can be seen to fit quite neatly into a family of shapes that correlate with that single parameter. This family is depicted in figure 1. These geometries reflect the fact that an axial flow machine, whether a pump, turbine, or compressor, is more efficient at high specific speeds (high flow rate, low head) whereas a radial machine, that uses the centrifugal effect, is more efficient at low specific speeds (low flow rate, high head). The existence of this parametric family of designs has emerged almost exclusively as a result of trial and error.

Normally turbomachines are designed to have their maximum efficiency at the design specific speed, N_D.

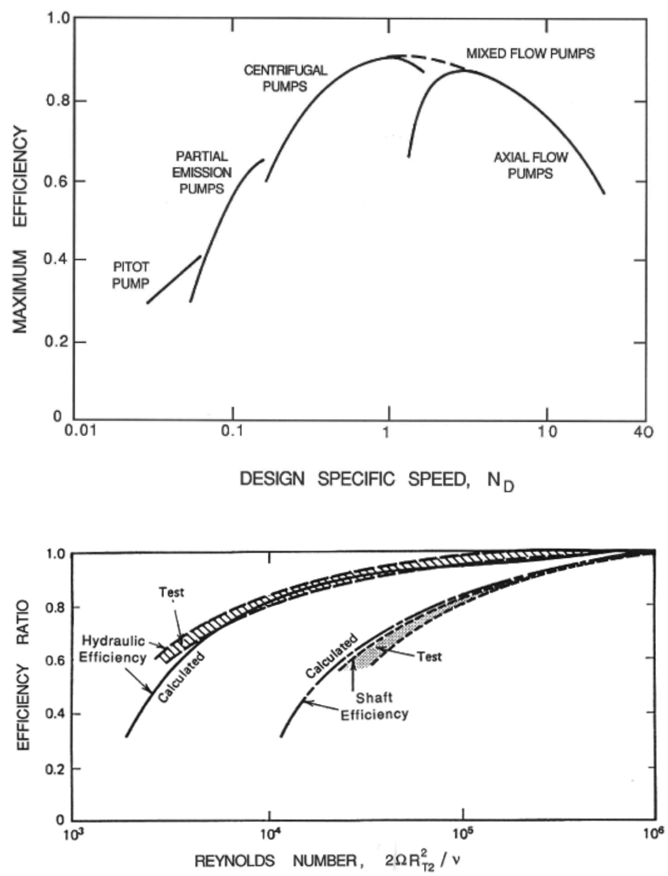

Figure 2: Compilation by Balje (1981) of maximum efficiencies for various kinds of pumps as a function of design specific speed, N_D. Since efficiency is also a function of Reynolds number the data has been corrected to a Reynolds number, $2\Omega R_{T2}^2/\nu$, of 10^8, using the correlation in the lower figure which shows the hydraulic efficiency, η_P, and shaft efficiency, η_S, as functions of that Reynolds number (from Balje 1981).

Thus, in any graph of efficiency against specific speed, each pump geometry will trace out a curve with a maximum at its optimum specific speed, as illustrated by the individual curves in figure 2. Furthermore, Balje (1981) has made note of another interesting feature of this family of curves in the graph of efficiency against specific speed. First, he corrects the curves for the different viscous effects which can occur in machines of different size and speed, by comparing the data on efficiency at the same effective Reynolds number using the diagram reproduced in figure 2. Then, as can be seen in figure 2, the family of curves for the efficiency of different types of machines has an upper envelope with a maximum at a specific speed of unity. Maximum possible efficiencies decline for values of N_D greater or less than unity. Thus the "ideal" pump would seem to be that with a design specific speed of unity, and the maximum obtainable efficiency seems to be greatest at this specific speed. Fortunately, from a design point of view, one of the specifications has some flexibility, namely the shaft speed, Ω. Though the desired flow rate and head rise are usually fixed, it may be possible to choose the drive motor to turn at a speed, Ω, which brings the design specific speed close to the optimum value of unity.