# Turbomachines

The term *turbomachine* refers to that class of fluid machines in which the fluid flow proceeding through the machine has its characteristics altered by a device (often called an impeller or a runner) that rotates within the fluid so that it either extracts energy from the flow or inserts energy into the flow. If energy is extracted to drive a generator or other device the machine is called a turbine; if energy is inserted using a motor the machine is called a pump or compressor. Turbomachines serve many purposes, come in a wide array of shapes and sizes and have a range of designs as will be seen in the sections which follow.

Though the constraints on a turbomachine design are as varied as the almost innumerable applications, there are a number of ubiquitous trends which allow us to draw some fairly general conclusions. To do so we make use of the affinity laws that are a consequence of dimensional analysis, and relate performance characteristics to the density of the fluid, $\rho$, the typical rotational speed, $\Omega$, and the typical diameter, $D$, of the turbomachines. Thus the volume flow rate through the turbomachine, $Q$, the total head rise or drop across the turbomachine, $H$, the torque, $T$, and the power absorbed or generated, $P$, will scale according to

$$Q \quad \alpha \quad \Omega D^3 \tag{Mac1}$$

$$H \quad \alpha \quad \Omega^2 D^2 \tag{Mac2}$$

$$T \quad \alpha \quad \rho D^5 \Omega^2 \tag{Mac3}$$

$$P \quad \alpha \quad \rho D^5 \Omega^3 \tag{Mac4}$$

These simple relations allow basic scaling predictions and initial design estimates. Furthermore, they permit consideration of optimal characteristics, such as the power density which, according to the above, should scale like $\rho D^2 \Omega^3$.

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 use a shaft rotating at a frequency, $\Omega$ (in $rad/s$), to accept a certain flow rate, $Q$ (in $m^3/s$) and to produce or utilize a head difference, $H$ (in $m$), between the flow entering and leaving the machine. 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:

$$N \;=\; \frac{\Omega Q^{\frac{1}{2}}}{(gH)^{\frac{3}{4}}} \tag{Mac5}$$

Though originally constructed to allow evaluation of the shaft speed needed to produce 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 change 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.

The above definition of the specific speed has employed a consistent set of units, so that $N$ is 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 later sections). 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 this monograph, 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.

Another typical consideration arising out of the affinity laws relates to optimizing the design of a turbomachine for a particular power level, $P$, and a particular fluid, $\rho$. This fixes the value of $D^5\Omega^3$. If one wished to make the turbomachine as small as possible (small $D$) to reduce weight (as is critical, for example, in the rocket engine pumps) or to reduce cost, this would dictate not only a higher rotational speed, $\Omega$, but also a higher impeller tip speed, $\Omega D/2$. However, as we shall see in a section that follows, the propensity for cavitation in a liquid turbomachine increases as a parameter called the cavitation number decreases, and the cavitation number is inversely proportional to the square of the tip speed or $\Omega^2 D^2/4$. Consequently, the increase in tip speed suggested above could lead to a cavitation problem. Often, therefore, one designs the smallest turbomachine that will still operate without cavitation, and this implies a particular size and speed for the device.

Furthermore, as previously mentioned, the typical fluid-induced stresses in the structure will be given by $\rho\Omega^2 D^4/\tau^2$, and, if $D^5\Omega^3$ is fixed and if one maintains the same geometry, $D/\tau$, then the stresses will increase like $D^{-4/3}$ as the size, $D$, is decreased. Consequently, fluid/structure interaction problems will increase. To counteract this the blades of the impeller or runner are often made thicker ($D/\tau$ is decreased), but this usually leads to a decrease in the hydraulic performance of the turbomachine. Consequently an optimal design often requires a balanced compromise between hydraulic and structural requirements. Rarely does one encounter a design in which this compromise is optimal.

Of course, the design of a pump, compressor or turbine involves many factors other than the technical issues discussed above. Many compromises and engineering judgments must be made based on constraints such as cost, reliability and the expected life of a machine. These sections will not attempt to deal with such complex issues, but will simply focus on the advances in the technical data base associated with cavitation and unsteady flows. For a broader perspective on the design issues, the reader is referred to engineering texts such as those listed at the end of this chapter.