Viscous Effects in Radial Flows

We now turn to a discussion of the viscous effects in centrifugal pumps. Clearly a radial cascade will experience viscous boundary layers on the blades that are similar to those discussed earlier for axial flow machines (see section (Mbcd)). However, two complicating factors tend to generate loss mechanisms that are considerably more complicated. These two factors are flow separation and secondary flow.

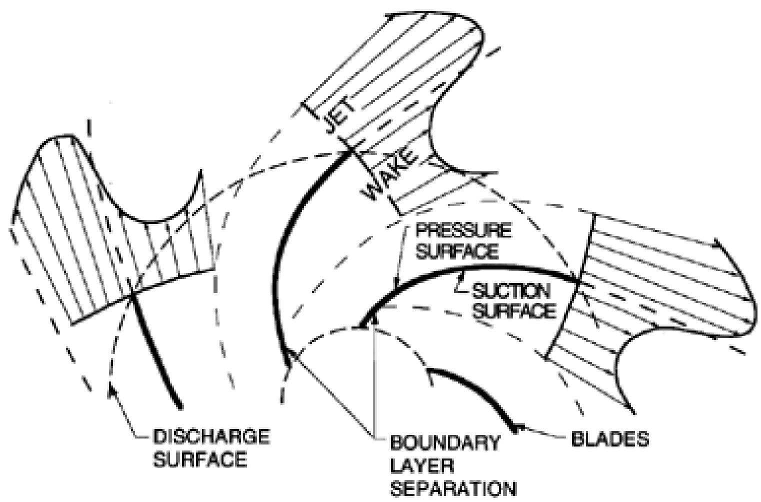

Figure 1: A sketch of actual discharge flow from a centrifugal pump or compressor including the alternating pattern of jets and wakes resulting from flow separation from the suction surfaces.

Normally, the flow in a centrifugal pump separates from the suction surface near the leading edge, and produces a substantial wake on the suction surfaces of each of the blades. Fischer and Thoma (1932) first identified this phenomenon, and observed that the wake can occur even at design flow. Normally, it extends all the way to the impeller discharge. Consequently, the discharge flow consists of a low velocity zone or wake next to the suction surface, and, necessarily, a flow of increased velocity in the rest of the blade passage. This “jet-wake structure” of the discharge is sketched in figure 1. Note that this viscous effect tends to counteract the displacement flow of figure 2, section (Mbce). Since the work of Fischer and Thoma, many others have studied this aspect of flows in centrifugal pumps and compressors (see, for example, Acosta and Bowerman 1957, Johnston and Dean 1966, Eckardt 1976), and it is now recognized as essential to take these features into account in constructing any model of the flow in radial turbomachines. Modern analyses of the flow in radial turbomachines usually incorporate the basic features of the jet-wake structure in the blade passages (for example, Sturge and Cumpsty 1975, Howard and Osborne 1977). Sturge and Cumpsty have calculated the shape of the wake in a typical, two-dimensional radial cascade, using numerical methods to solve a free streamline problem similar to those discussed in section (Mbes).

At design flow, the wake or boundary layer on the suction surface may be quite thin, but as the flow coefficient, ϕ, is decreased, the increased incidence leads to larger wakes (Fischer and Thoma 1932, Johnston and Dean 1966). Clearly, the nonuniformity of the discharge flow implies an “effective” slip due to these viscous effects. This slip will not only depend on the geometry of the blades but will also be a function of the flow coefficient and the Reynolds number. The change with flow coefficient is particularly interesting. As ϕ is decreased below the design value and the wake grows in width, an increasing fraction of the flow is concentrated in the jet. Johnston and Dean (1966) showed that this results in a flow that *more* closely follows the geometry of the pressure surface, and, therefore, to a *decrease* in the slip. This can be a major effect in radial compressors. Johnston and Dean made measurements in an 18-bladed radial compressor impeller with a 90° discharge blade angle (for which $Sf_S = 0.825$), and found that the effective slip factor increased monotonically from a value of about 0.8 at $\phi_2 = 0.5$ to a value of 1.0 at $\phi_2 = 0.15$. However, this increase in the slip factor did not produce an increase in the head rise, because the increase in the viscous losses was greater than the potential gain from the decrease in the slip.

Finally, it is important to recognize that secondary flows can also have a substantial effect on the development of the blade wakes, and, therefore, on the jet-wake structure. Moreover, the geometric differences between the typical radial compressor and the typical centrifugal pump can lead to significant differences in the secondary flows, the loss mechanisms, and the jet-wake structure. The typical centrifugal pump geometry was illustrated in sections (Mbbb) and (Mbbe), to which we should append the typical number of blades, $Z_R = 8$. A typical example is the geometry at $N_D = 0.6$, namely $R_{T1}/R_{T2} \approx 0.5$ and $B_2 \approx 0.2R_{T2}$. Assuming $Z_R = 8$ and a typical blade angle at discharge of 25°, it follows that the blade passage flow at discharge has cross-sectional dimensions normal to the relative velocity vector of $0.2R_{T2} \times 0.3R_{T2}$, while the length of the blade passage is approximately $1.2R_{T2}$. Thus the blade passage is fairly wide relative to its length. In contrast, the typical radial compressor has a much smaller value of B_2/R_{T2}, and a much larger number of blades. As a result, not only is the blade passage much narrower relative to its length, but also the typical cross-section of the discharge flow is far from square, being significantly narrower in the axial direction. The viscous boundary layers on the suction and pressure surfaces of the blades, and on the hub and shroud (or casing), will have a greater effect the smaller the cross-sectional dimensions of the blade passage are relative to its length. Moreover, the secondary flows that occur in the corners of this passage amplify these viscous effects. Consequently, the flow that discharges from a blade passage of a typical radial compressor is more radically altered by these viscous effects than the flow discharging from a typical centrifugal pump.