tant in establishing the observed flow. The unshrouded impeller showed the strong influence of the flow entering the passage due to the clearance between the blade tips and the stationary shroud. The strongest vorticity was near the tip but it spread into the midpassage toward the impeller side. A large single vortex was established which filled the passage. Viscous effects on the hub surface accounted for the vorticity in the opposite direction close to the hub.

Despite the considerable change in the secondary flow pattern between the shrouded and unshrouded versions, it appears that the average primary velocity profile was basically unaltered. However, some disturbances in the unshrouded impeller can be noted near the blade tip in Figs. 9 to 12, which lead, at the passage exit, to decreasing radial velocities and increasing slip angles toward the tip.

Conclusions

Measurements were carried out which defined the distribution of primary and cross-passage velocity components within a rotating radial impeller passage for shrouded and unshrouded configurations.

The primary flow pattern was stable with no tendency toward separation of the suction side boundary layer even near the passage exit. This flow pattern was generally maintained for both configurations, with minor variations such as a locally reduced velocity near the blade tip, in the unshrouded impeller leading to increased slip in that region.

The shrouded impeller had a secondary flow pattern, shown by the distribution of cross-passage velocity, which progressed from a single vortex at entry to a vortex near each of the hub and shroud surfaces. The backward curvature of the streamlines through the passage was nearly balanced by the Coriolis force so that the entry velocity profile had a weak influence in the secondary flow generation.

The secondary flow pattern measured in the unshrouded impeller passage appeared to be very strongly influenced by the flow entering the passage through the clearance between blade tip and shroud. A single vortex soon filled the passage. It is likely that a passage geometry which had a stronger capability to generate secondary vorticity from upstream velocity gradients, would have its secondary flow pattern modified to a lesser degree.

Any attempt to predict the three-dimensional flow field in an impeller should include a correct model for the secondary flow pattern. A calculation method based on inviscid flow, such as that employed in [2], is unable to predict the pattern measured in the unshrouded impeller. In some cases, it might be useful to note that the flow pattern is qualitatively the same as that in one-half of a shrouded passage of twice the depth, where the entry flow has strong hub and shroud boundary layers. A complete analysis of the three-dimensional flow field in an impeller, particularly an unshrouded one, should be able to account for local viscous effects and cross-passage velocity injection. One calculation method which sets out to include these capabilities for a general passage flow is that of Patankar and Spalding [7], which has, however, not been applied to an impeller passage and may be unable to handle boundary layer separation and passage exit effects.

References


Why do some radial impellers exhibit separation while others avoid separation? This may be understood by considering the radial impeller passage as a rotating diffuser. Like the stationary diffusing passage, the fluid dynamic performance of the rotating diffuser depends upon geometry and inlet flow conditions as well as rotational effects. We know that for the stationary diffuser the flow can remain uninstalled if the area ratio is small and the length-to-inlet hydraulic diameter ratio is sufficiently large. Recent measurements by Rothe and Johnston [10] on rotating, plane wall diffusers have clarified the flow behavior of rotating diffuser performance. It is clear from their data that rotational effects reduce the area ratio at which a given configuration will stall and that this reduction in "area ratio to first stall" is a gradual effect dependent upon the inlet flow conditions and rate of rotation. However, for sufficiently small area ratio, a diffuser can remain uninstalled even for large rates of rotation.

For diffuser geometries somewhat similar to that used in the paper (although plane wall and with fully developed inlet boundary layers), Rothe and Johnston found the line of first stall to follow the trend shown in Fig. 18. AR is the overall diffuser area ratio and R0 = ωW/D. From Fig. 2 of the paper, the impeller passage area ratio has

DISCUSSION

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Professor Howard and Mr. Kittmer have presented an excellent documentation of the relative velocity field inside a rotating centrifugal impeller. This type of information will be needed to comprehend the fluid mechanics of radial machines if improved design and performance prediction techniques are to be evolved.

The authors' data suggest a potential-like flow altered by boundary layer growth on the hub, shroud, suction, and pressure surfaces. The secondary flow patterns observed apparently arise from passage curvature, Coriolis accelerations and, in the case of a shrouded passage of twice the depth, where the entry flow has strong hub and shroud boundary layers. A complete analysis of the three-dimensional flow field in an impeller, particularly an unshrouded one, should be able to account for local viscous effects and cross-passage velocity injection. One calculation method which sets out to include these capabilities for a general passage flow is that of Patankar and Spalding [7], which has, however, not been applied to an impeller passage and may be unable to handle boundary layer separation and passage exit effects.


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5 Numbers in brackets designate additional References at end of Discussion.
been estimated at approximately 1.14 and this area ratio is shown in Fig. 18. It is believed that the rotation number $R_0$ should more properly be based on the inlet boundary layer thickness rather than the inlet width (Rothe and Johnston's data had fully developed inlet flow and hence the boundary layer thickness was approximately one-half the throat width). Using a crude estimate of a boundary layer thickness of 1/8 in. near the inlet of the impeller, the rotation number $R_0 = \frac{228}{U_1} = 0.04$ for the authors' data. Thus the particular impeller flow geometry of the paper, when examined as a rotating diffuser, could be expected to fall well below the line of first stall. Of course in the radial machine, the simple correlation shown in Fig. 18 will be influenced by other factors such as passage curvature, blade back sweep and boundary layer history effects. Therefore the line of first stall may be expected to be different than that shown in Fig. 18, although we suspect the small area ratio used by the authors to still be below first stall.

If the same arguments are applied to high pressure ratio centrifugal compressors, it is found that the rotation number and effective passage area ratios of interest in practical designs almost always lie above the line of first stall. Thus these machines should have separated jet/wake type flows in the impeller.

Finally, the authors may wish to compare their measured secondary flow patterns with those observed and reported in [11] in a photographic study of a centrifugal pump.

**References**


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**Authors’ Closure**

Dr. Runstadler has introduced a potentially valuable tool for the classification of the flow regimes in centrifugal impeller passages. To assist in its evaluation, two further estimated operating points have been added to the discusser’s Fig. 18 (see the preceding figure). They are a backward-curved radial impeller tested by Lennemann (paper references [1, 2, 3]) and a rotating straight radial diffuser (Fowler). The $R_o$ values were estimated for Lennemann’s impeller, assuming a 1/8 in. entry boundary layer thickness and for Fowler’s, choosing the full flow conditions and assuming entry boundary layers one-tenth the channel width. Both the Lennemann and Fowler impellers, which lie above the first stall line, had exit velocity profiles decreasing toward the suction surface, in contrast to the opposite “potential-like” velocity gradient shown in the present study. It is interesting to note that all three cases lie below the first stall line for (their particular length/width ratio) on a Kline map for stationary two-dimensional diffusers. At this time, the criteria should be used with caution, even for radial impellers, since Rothe and Johnston’s data were established for a diffuser length/width ratio of 6 and the effects of passage curvature and aspect ratio are unknown.

The secondary flow pattern shown by Senoo (discusser’s reference [4]) is more complex than for the unshrouded case in the present study. The tip clearance flow appears to modify only a small portion of the impeller passage secondary flow. His mixed-flow impeller has a sharp bend from axial and deep radial passages which probably generate strong cross flows. We have just seen the report of a flow visualization study of an impeller with straight radial blades where, in common with our measurements, a twin-vortex pattern was observed in a shrouded version of the impeller and a single vortex in an unshrouded impeller.