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impeller is stalled and rotating stall observed at that condition is attributed to the impeller. In Fig. 10, those data that are judged as impeller stall based on the shape of the characteristic curves are indicated as triangular marks.

According to [10], rotating stall induced in the impeller contains strong harmonics in the velocity signal while rotating stall induced in the vaneless diffuser has an isolated frequency and weak harmonics; consequently, the signal may be used to judge the source of rotating stall. Applying this criterion to the published pressure signals in [4, 5], numbers 10, 12, and 14 in Fig. 10 are suspected as rotating stall induced in the impeller, and they are indicated as square marks.

Number 17 in Fig. 10 was once regarded [8] as the diffuser stall condition, but later experiments were repeated [10], and the new result is indicated as number 17′ in Fig. 10. Probably the data number 17 should be replaced by the data number 17′.

In cases of multistage centrifugal compressors with very narrow vaneless diffusers, the shaft diameter is made large to increase the critical speed. As a result, the inlet eye diameter of the impeller is unduely large as a low specific speed impeller. At a small flow rate the inlet relative velocity approaches the circumferential velocity of the blade-leading edge, which is about one-half of the tip speed for these impellers. On the other hand, since the mean relative velocity inside the impeller is approximately proportional to the flow rate, it is very small compared with the inlet relative velocity when the flow rate is little or when the absolute flow angle at the impeller exit is small. Because of the unduely large deceleration of the relative velocity in the impeller, it is very likely that the flow separates from the blade surface and the impeller is stalled.

If an impeller has blades with large backward leaning angle, the pressure versus flow rate characteristic curve may keep a negative gradient, even if the impeller is stalled at a small flow rate and the performance of the impeller is somewhat deteriorated. As a result, it is usually possible to operate the compressor stably, including the stall condition of the impeller. Because of the stable characteristics of the impeller, in cases of industrial centrifugal compressors an impeller designed for a large flow rate is sometimes combined with a narrow vaneless diffuser. In these cases, the impeller-blades are stalled at a flow coefficient which is not so small for the diffuser. In cases of numbers 5, 12, 13, 14, 15, and 16 in Fig. 10, the diffuser is narrower than the exit width of the impeller, and it may be possible that rotating stall occurred in the impeller before it does in the vaneless diffuser.

At least the data indicated in Fig. 10 as open circles were obtained under the acceptable condition, and the prediction indicated in the figure as a full line agrees well with those experimental data. When a vaneless diffuser is followed by a scroll or a collector as the last stage of a multistage compressor, the flow is not axisymmetric at an off-design condition, and the critical flow angle may be changed. This problem is left for a future study.

Conclusions

1 In the case of a very narrow diffuser with b/r_i = 0.0153, rotating stall was not observed until the mean inlet flow angle was reduced to 3.4 deg from the circumferential direction as predicted.

2 The critical flow angle for rotating stall was influenced little by modification of the exit/inlet radius ratio in the tested very narrow vaneless diffuser.

3 The critical inlet flow angle can be predicted correctly for very narrow diffusers as well as for wide diffusers.

4 The propagating speed of stall cells induced in vaneless diffusers agreed fairly well with the prediction [1].

5 In cases of small specific speed blowers, the impeller may stall before the vaneless diffuser stalls unless a special care is taken for designing the impeller.

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DISCUSSION

R. Van den Braembussche[1]

The authors have made several important contributions to a better understanding of the flow in vaneless diffusers and the data presented in this paper are an additional step in the confirmation of their theoretical prediction, for which they must be congratulated.

However, I want to comment on some of the experimental points which are left on Fig. 9. Most of the points on Fig. 10 have been refused because they should not correspond to diffuser rotating stall. Several arguments were used for this. Points 8, 11, 13, 15, 17, 18, and 19, indicated by an open triangle, are refused because the critical flow angle corresponds to a kink in the pressure rise curve or to peak pressure rise. However, the performance curve corresponding to point 6 (Fig. 4(*a*) of [3]) and point 9 (Fig. 7 of [3]) also show a kink in the pressure rise curve at critical flow. It is not clear why these points are left on Fig. 9, and if they are also taken off, only three points are left to compare with the theory.

Measurements of Abdelhamid [14] have shown that diffuser rotating stall can result in a kink in the performance curve (Fig. 3 of [14]) and that these instabilities can be removed by a throttling of the diffuser outlet. We are therefore not convinced that a kink in the pressure rise curve is a sufficient condition to conclude for impeller rotating stall.

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At the time we made the comparison shown on Fig. 10, there were not many criteria available to distinguish between impeller and diffuser rotating stall. We therefore based our judgments on the published information and our own observation. However, we still think that there are many arguments to accept that the unstable conditions given on that figure are due to diffuser rotating stall.

For points 1, 2, 3, and 5 rotating stall is observed on the negative slope side of the pressure rise curve without any discontinuity.

For points 2 and 5 the rotational speed of the cells is in agreement with the theory of Jansen [1]. This could not be checked for all the points because of an uncertainty in the number of stall cells.

Impeller rotating stall has a strong upstream influence. A hot film probe installed in the inlet channel of point 5 did not show any low-frequency perturbation before surge occurred.

Points 1, 2, 3, and 5 are obtained with impellers with 65 to 68 deg backward lean angle, similar to point 9, for which it was mentioned in [3] that this would prevent the impeller from stalling.

The discrepancy between experiments and predicton of points 12, 13, 14 and 6, 7, 8 can be explained by diffuser inlet flow distortion. Each series of data is obtained with one impeller and three diffusers of different width. It is quite possible that the sudden contraction at diffuser inlet creates a diffuser inlet flow distortion. The theory [15] predicts a 3 to 7 deg change in critical flow angle due to inlet distortion.

For the other points the discrepancy between theory and experiments was explained in [8] as a Reynolds number influence, based on experimental observations. The pairs of data 1, 3 and 18, 19 are obtained with the same impeller and diffuser combination but at different Reynolds number, corresponding to a different pressure level. The theory does not predict a direct Reynolds number influence on diffuser stability but it could be that lower Reynolds numbers result in higher flow distortion at impeller exit, and therefore also influence diffuser stability.

One should not conclude from this that we have any doubt about the validity of this theory. On the contrary, the data shown in this paper prove that a very good agreement is obtained when the experimental diffuser inlet flow is close to uniform as assumed by the theory. As a conclusion, I would suggest that when using this prediction, one has to care more about diffuser inlet flow perturbations due to impeller flow, diffuser inlet geometry, or leakage flow.

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Author's Closure

Several reasons are conceivable for scattering of experimental data in Fig. 10. A distorted velocity distribution across the diffuser width at the inlet and a circumferentially distorted boundary condition at the diffuser exit due to the scroll are some of them. In addition, it is suspected that data of rotating stall in the impeller may be erroneously included.

In the process of deducing a relationship between variables regarding a few experimental data, a single erroneous datum can upset the whole relationship; on the other hand, the relationship is hardly misled by lack of a datum. Therefore, in the present research it is important to limit the experimental data to only very reliable ones and the data should be consistently presented.

At a small flow rate impeller blades must be stalled as it will be mentioned later. Stall of impeller blades does not always mean incipience of rotating stall in the impeller, but in many cases rotating stall is induced at the critical flow rate of impeller stall. Therefore, each of the data in Fig. 10 was examined to determine whether it was safe for impeller stall. If a datum satisfied one of conditions of possible impeller stall, it was not adopted in Fig. 9 unless the information on the experiment convinced the authors that the impeller was not stalled. The conditions of possible impeller stall are explained in the text for respective experimental points.

Rotating stall in a vaneless diffuser hardly changes the mean intensity of swirl in the diffuser unless local reverse flow occurs at the diffuser exit, and the pressure-flow rate characteristic curve does not have anomaly at the critical condition. Therefore, if anomaly is observed in the characteristic curve at the critical flow rate of rotating stall, it is at once suspected that the rotating stall is induced in the impeller. However, if the asymmetric reverse flow, which is induced by the rotating stall in the vaneless diffuser, reaches the diffuser exit, the mean pressure rise in the diffuser becomes less and anomaly appears in the characteristic curve. Therefore, at the critical condition where the characteristic curve has anomaly the incidence angle and the deceleration ratio at the inlet of the impeller are examined as a rule. If the impeller is safe for stall, it is assumed that the rotating stall is induced in the vaneless diffuser.

The mean velocity in the throat between blades at the inlet of the impeller is proportional to the flow rate. At a small flow rate the velocity is very low, and unless the circumferential velocity at the impeller inlet is very small, the deceleration rate from the impeller inlet to the throat is too large for the boundary layer to flow along the blade surface and rotating stall may be induced. However, an impeller with blades of large backward leaning angle has a characteristic curve with a steep negative gradient, and it can run stably even at the small flow rate and the impeller stall is not apparent in the performance.

In other words no impeller can work without separation of flow near the leading edge of the blades when the flow rate is reduced to less than 1/2 or 1/3 of the zero incidence flow. If a wide impeller is used for a test of a narrow vaneless diffuser, when the flow rate is reduced the impeller likely stalls before the rotating stall occurs in the vaneless diffuser. Therefore, the critical data of vaneless diffusers which were tested with an impeller wider than the diffuser width should be examined carefully to determine whether the impeller is the cause of rotating stall.

The difference in the critical flow angles of points 1 and 3 indicates the influence of a change of Reynolds number. The flow at the impeller exit and the flow in the vaneless diffuser are influenced by Reynolds number to a certain degree, but probably the effect of Reynolds number on the decelerating zone at the impeller inlet is the most serious, and at a low Reynolds number laminar separation and rotating stall in the impeller may result at a relatively large flow rate.

Points 18 and 19 show that the critical flow angle was modified by 4 deg by a change of Reynolds number, but according to [9], the critical flow coefficient was identical for the two cases. It is presumed that the velocity distribution was distorted axially or circumferentially depending upon the Reynolds number and that the flow angle was measured at the middle of the diffuser passage. In this paper data are handled on the basis of uniform inlet flow, and if the critical flow angles of points 18 and 19 are evaluated on this basis, the two points must be identical because of the identical flow coefficient.

Because of the above reasons, many of data in Fig. 10 were tentatively withheld in Fig. 9 which indicates the critical conditions of rotating stall in vaneless diffusers. If the experimental

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conditions are disclosed and it becomes clear that the impellers were not the cause of the rotating stall, those data should be adopted in Fig. 9.

It is hoped that experiments on rotating stall in vaneless diffusers in the future will be carried out using impellers which are safe for stall at a flow rate less than the critical flow rate of the vaneless diffusers, and the critical flow rate of the impeller should be clearly noted. Using those reliable data, the influences of various parameters on the critical flow angle of vaneless diffusers will be clarified.

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