Acknowledgments

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DISCUSSION

A. Whitfield[2] and F. J. Wallace[3]

The paper presented is of particular interest as we have been concerned with a similar exercise of attempting to identify and predict the individual loss components within a centrifugal compressor, with a view to obtaining theoretical compressor performance at both design and off-design conditions. The authors appear to use a very basic procedure in order to estimate the incidence loss H_{SH}. Due to the different inducer designs the incidence loss attributable to impellers *B, S* and *F* may differ considerably. We have studied two alternative incidence loss models[4] and developed them to account for the variation of incidence, approach relative Mach number, and blade thickness from hub to shroud. These variations may be particularly relevant to impellers *B* and *F* as, judging from Fig. 1 *(a),* the inducers do not appear to be constructed of radial blade fibers.

On the other hand, the authors have presented a detailed empirical procedure for the computation of the mixing loss, whereas at the University of Bath we have adopted a more direct procedure. We consider the magnitude of the throughflow jet to be a function of the degree of separation within the impeller. The amount of diffusion attained before separation cannot be predicted with a one-dimensional flow technique, indeed existing two and three-dimensional flow analysis still lacks the necessary boundary layer theory in order to predict separation. In view of this the technique used is simply to specify the diffusion ratio attained prior to separation, this then leads to a predicted jet and wake width and hence the mixing loss. We find that the specification of the impeller diffusion ratio is the most crucial parameter required in the performance prediction procedure. In view of this we wonder whether the continued search for more and more sophisticated empirical loss equations is really worthwhile. With their experience, how do the authors feel about this point?

At the University of Bath we are investigating and designing turbocharger compressor impellers with a view to improving the operating range between surge and choke. We are presently designing an impeller with backward swept blades, similar to impeller *B,* and are, therefore, concerned about the poor performance of impeller *B* shown in Fig. 4. Do the authors have any explanation as to why the performance curve is terminated by an early surge, as this is contrary to general experience with backward swept vanes?

Finally, the definition of the Reynolds number Re_{wl} needs clarifying. In the Nomenclature it is based on impeller exit conditions w_2 and ν_2 whilst on p. 3 it is stated to be based on the impeller inlet conditions.

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[4] Whitfield, A., and Wallace, F. J., "Study of Incidence Loss Models in Radial and Mixed Flow Turbomachinery," IMechE Conference, Publication 3, Paper No C55/73, 1973.

A. C. Bryans.[5]

Professor Watanabe, et al., have given us a very comprehensive survey of the relative influence of those intrinsic mechanisms which contribute to the propagation of loss in a low pressure compressor stage. It would be interesting to see similar results in the 4:1 and 6:1 pressure ratio range. We would surmise that as we would move to higher impeller pressure ratios the influence of Reynolds number would be much more significant in the impeller than in the diffuser.

Turning to the relative level of the losses in Table 4, we note H_1, H_{df}, H_s, and H_e are consistently the highest items in each configuration. H_1 is due to the leakage through the impeller/shroud gap. This gap of 0.4 mm is about double what we would normally expect to hold. So we would conclude that leakage loss is too high. H_{df}, the diffuser loss, is very much dependent on the condition of the fluid as it enters the diffuser. This, in turn, is intimately related to H_{mix} and the "jet-wake" nature of the impeller exit flow. We suspect that this is a powerful contributor to the magnitude of H_{df}. H_e is merely dependent on design configuration and could virtually be eliminated. H_s and H_{mix} are probably the most bothersome,—they are functions of blade surface angle, blade number, and meridional flowpath contour. It would require considerable effort to unravel purely configuration influences from direct Reynolds number effects. The heuristic approach of the authors is to be commended and it is hoped that further investigations of this type will be forthcoming.

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Authors' Closure

The present authors wish to express appreciation to Dr. A. C. Bryans, Dr. A. Whitfield, and Prof. F. J. Wallace for their pertinent comments and suggestions.

As to the queries by Dr. A. Whitfield and Prof. F. J. Wallace, the authors' answer is as follows.

It is surely true that the diffusion ratio is the crucial parameter if the separation induced within a rotating impeller channel is equivalent to that generated within a stationary diffuser. The authors, however, consider that the triggers to the separation of the flow within the rotating impeller channel are divided into two kinds. The first one is the aforementioned separation which would be generated within a stationary divergent flow channel as diffuser. The other is one which would be induced by the coriolis force. The latter may be equivalent to the separation which would be risen from the inside wall by a bend of duct (refer to Fig. 9). Furthermore, it would be conceivable that so far as the flow within the rotating impeller channel is concerned, the separation caused by the Coriolis force will have more significant effect on width of the wake than that induced by the diffusion. Thus, the authors adopted equation (14) for the evaluations of the width of the wake.

Concerning an early surge of the impeller *B*, the affairs are as follows.

In case of impeller *S* in Fig. 4, the flow rate $G\sqrt{T_0}/P_0$ corresponding to the point located utmost left on each constant rotational speed curve coincides approximately with those just before surge. The measured points for impellers *D*, *B* and *F* are merely selected to compare their performance characteristics under the same operational conditions, respectively. Thus, the flow rate ($G\sqrt{T_0}/P_0 = 3.9 \times 10^{-4}$) corresponding to the point utmost left on the performance curves for impeller *D* and *B* does not imply the surge point. On the other hand, as to impeller *F*, the point utmost left coincides with the surge point. This will be confirmed by the column in Table 4 corresponding to the operating conditions. As is shown in case of impeller *F*, the minimum flow rate which could be measured under $u_2/\sqrt{T_0} = 11.6$ has been finished at $G\sqrt{T_0}/P_0 = 4.4 \times 10^{-4}$, because of the appearance of surge. On the other hand, in case of impellers *S*, *D*, and *B*, the measurements under operational condition of $G\sqrt{T_0}/P_0 = 3.9 \times 10^{-4}$ could be achieved.

Furthermore, Dr. Whitfield and Professor Wallace pointed out that impeller *B* had a poor performance. This is valid so far as the pressure ratio is concerned, because the ideal head obtained by the impellers rotating at the same rotational speed is the largest in case of impeller *F* and the smallest in case of impeller *B*. On the other hand, if the adiabatic efficiency ψ_4/μ are concerned, the maximum efficiencies are the highest in case of impeller *B* (refer to Fig. 10).

Finally, as to the definition of the Reynolds number Re_{w1} the subscript 1 is a misprint. Thus, the subscript 1 has to be replaced by *l* denoting flow path length.

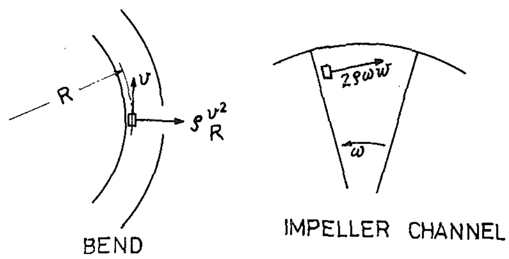

Fig. 9

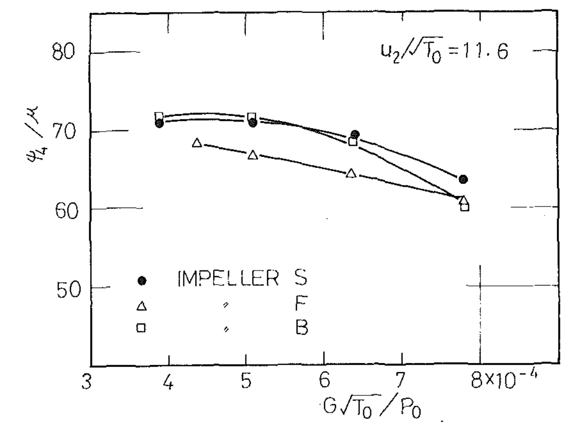

Fig. 10