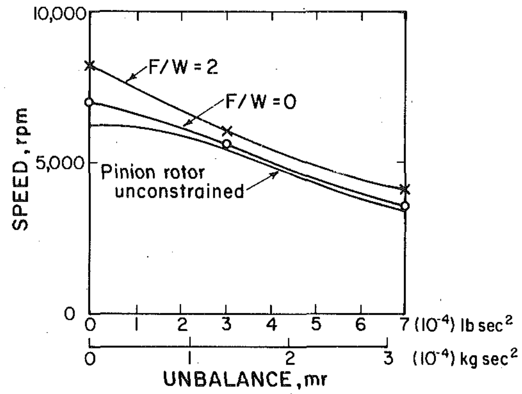

Fig. 4 Threshold of instability of gear system for gear ratio 1:1. *x*—instability occurs first in pinion rotor, O—instability occurs in both rotors at the same time

position = 0.0005 in. = 0.00127 cm. The oil used is SAE 10 at 150°F (65.5°C) average temperature (corresponding to an average viscosity of 1.76×10^{-6} Reynolds).

The numerical solution for the eccentricity ratios and its range of fluctuation at different speeds is given in Figs. 2 and 3 for gear ratios of 1 and 2, respectively. The calculated data are for transmitted gear loads F equal to zero and $2W$ where W is the pinion rotor weight.

The results for the unconstrained pinion rotor are plotted in Fig. 2(a) for comparison.

It can be seen from the figures that the transmitted gear load has, as expected, a stabilizing effect on the gear (since its effect is additive to its static weight) and a destabilizing effect on the pinion (since its effect is to reduce the static load on the bearings).

The example of identical gears and pinions with identical unbalance magnitude is believed to illustrate the transmitted load effects on the total system stability. The results for the threshold of instability are plotted in Fig. 4. As can be seen, the gear constraint with zero transmitted load causes an increase in the threshold speed. Instability in this case occurs, as expected, identically in the pinion and the gear rotors. The effect of the transmitted load can be seen to increase the threshold of instability in this case with the pinion rotor instability occurring first and therefore causing the system instability.

References

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Journal of Engineering for Power Discussion

Effects of Non-uniform Passages on Deepfold Heat Exchanger Performance[1]

A. L. London.[2] This paper deals with a difficult question: namely, how to specify manufacturing tolerances for laminar flow type heat exchanger surfaces. The discussor became aware of this problem in 1965, when he was at the General Motors Research Laboratory working with Mr. Mondt. Two papers in 12 years on this open question is possibly characteristic of the response of engineering technology to nagging, as distinct from demanding, problems. The author is to be congratulated for an excellent contribution. With like contributions, the designer will eventually have the information he needs to specify tolerances.

The task will remain a formidable one, especially until operationally convenient methods of measurement of the deviations such as fin-spacing and bulginess are developed. In this respect, the presentation in Appendix B is all too brief. It would be worthwhile to have Mr. Mondt expand on this subject in another paper.

The idea of combining a bulginess characterization with a fin-spacing characterization into a single total deviation characterization is appealing. Unfortunately, equation (10) for σ_t^* does not obviously combine equation (5) for σ_c^* with equation (9) for σ_b^*. A more direct formulation might be

$$\sigma_t^{**} \triangleq \sqrt{\sigma_c^{*2} + \sigma_b^{*2}} \qquad (17)$$

This total deviation characterization differs from the equation (10) definition of σ_t^*. Numerically, however, using the Table 1 standard shapes, σ_t^{**} and σ_t^* agree within six percent. The test packs of Table 2 show a larger difference ranging up to 16 percent. Unfortunately, the experimental uncertainties in the test data (Figs. 10, 11) are such that one cannot tell whether σ_t^* or σ_t^{**} provides the better correlation with the theory curve.

A question arises in the definition of the authors σ_t^*, equations (10) and (11). Does this expression characterize both bulginess (σ_b^*) and fin nonuniform spacing (σ_c^*)? If $\sigma_b^* \to 0$, σ_t^* should go to σ_c^*. If $\sigma_c^* \to 0$, σ_t^* should go to σ_b^*. This is the case for $\sigma_b^* \to 0$, but not for $\sigma_c^* \to 0$. However, if σ_b^* is redefined from the authors equation (9) to

$$\sigma_b^{**} = \sqrt{\frac{1}{(n-1)} \sum_{1}^{n} \left[\frac{b_i}{c_i} - 1\right]^2} \qquad (18)$$

then σ_t^* of equations (10) and (11) would "behave" for both limiting cases.

A research task remains which requires more precise experimentation as well as more sophisticated statistical techniques for its resolution. In the meanwhile, Mr. Mondt's paper serves as a reminder to industry that this is a nagging problem that won't go away. It needs to be resolved!

[1] By J. R. Mondt, published in the October 1977 issue of the ASME JOURNAL OF ENGINEERING FOR POWER, Vol. 99, No. 4, pp. 657–663.

[2] Professor, Department of Mechanical Engineering, Standford University, Stanford, CA.

Author's Closure

The author acknowledges with gratitude the timely discussion by Professor A. L. London. Professor London suggests a slightly different definition for bulginess, σ_b, and a different definition for combined total nonuniformity, σ_t. The proposed total combined nonuniformity, equation (17), is the root-sum-square of the bulginess and nonuniform fin spacing, σ_c, in contrast to the author's equation (10).

The author agrees that the root-sum-square approach appears more rigorous since σ_t collapses to σ_b when σ_c approaches zero. In fact, the root-sum-square approach was initially attempted by the author. However, the passage-scanning software and statistical calculations exceeded the storage capacity of the computer facility which was available to the author during the study. By using the shorter calculation procedure, outlined in the paper, a sample of 240 passages could be used for the statistical calculations. This large sample size was chosen to assure with 95 percent confidence that both the mean fin spacing and bulginess as well as the corresponding standard deviations for the passage population were estimated within ±0.5 percent by the sample measurements.

Perhaps some combinations of nonuniform geometries correlate better with equation (10), and some correlate better with equation (17). Therefore, the next logical step, if this work is extended, would be to compare nonuniformity parameters calculated by both equations using measurements from several typical nonuniform geometries.

Periodically Unsteady Flow in an Imbedded Stage of a Multistage, Axial-Flow Turbomachine[1]

G. J. Walker.[2] The authors are to be complimented on their detailed three-dimensional description of the unsteady velocity field through a compressor stage. The orderly behavior of the wake transport processes observed at different radial positions encourages the belief that allowances for wake interaction effects will ultimately be incorporated in axial turbomachinery design.

The use of helium injection to trace wake flows in an axial machine has been reported previously by Kerrebrock and Micolajczak [10]. The present paper now demonstrates that heated fluid can also be used to advantage in locating the IGV wake transport through a compressor. Further use of such passive marker techniques will undoubtedly be necessary to reveal the fine structure of the various wake interaction processes.

The 50 percent span rotor exit periodic-average axial velocity profiles described in the present paper (and shown in Fig. 6 of Schmidt and Okiishi [6]) appear quite consistent with the earlier more qualitative observations made in a similar situation by Lockhart and Walker [12], and shown in Fig. 3 of that paper. (There are, of course, some differences in appearance of the two sets of observations due to the latter measurements being offectively a plot of streamwise velocity along a streamwise traverse line, which cuts the various wake regions at a different angle). However, the Tasmanian work [12], in which observations were made at several axial stations between blade rows, suggests that some aspects of the present measurements can be interpreted in a rather different manner. For this reason some further discussion of wake-wake interaction phenomena seems warranted.

There is general agreement about the wake chopping and transport as represented in the periodic-average cascade plots of the present paper (Fig. 4 for the data under discussion). The IGV wake-rotor wake interaction is initiated by this chopping and the influence of IGV wake segments adjacent to the rotor wake. Slip flow within the IGV wake will tend to thin the rotor wake on the pressure side of the IGV wake street due to both flow convection and turbulent mixing effects; this thinning effectively restricts the slip flow towards the rotor trailing edge in the rotor wake, so that low energy rotor wake fluid accumulates against the suction side of the IGV wake street. The local rotor wake thickening thus produced is accentuated by the convective effects of slip flow within the IGV wake segments adjacent to the suction side. This process commences immediately downstream of the rotor trailing edge, and with increasing distance downstream there is a progressive thickening of the rotor wake on the suction side of the IGV wake street and a thinning on the pressure side (as indicated by the measurements in Fig. 4 of [12]). It is true that low energy IGV wake fluid must accumulate on the pressure surface during passage through the rotor, as noted by Kerrebrock and Mikolajczak [10]; but the IGV wake velocity defect is relatively small at the rotor exit, and cannot produce continued streamwise variations downstream of the rotor of the magnitude observed in [12]. According to this reasoning, the the deepest periodic-average rotor wake observed at 50 percent span for $YO_R/S_R = 0.69$ is interpreted as being mainly due to accumulation of rotor wake slip flow; the rotor wake region is in fact thicker here than on the pressure side of the IGV wake street, as shown in the profile for $YO_R/S_R = 0.17$.

The accumulation of rotor wake slip flow against the suction side of the IGV wake street immediately downstream of the rotor means that intra-stator rotor wake-stator interactions as described by Kerrebrock and Mikolajczak [10] will generally be greatly diminished for an imbedded stage. The circumferential location of the major rotor wake fluid accumulation will then be controlled mainly by the circumferential position of the next upstream stator row, and will not in general coincide with the stator pressure surface.

It is noted that Fig. 4 of [10] shows results of helium tracer tests which do indeed indicate the accumulation of rotor wake fluid on the stator pressure side in a compressor with IGV, rotor and stator rows; but the above discussion suggests that this should be regarded as a chance result in an isolated test[3], Lockhart and Walker [12] found that the region of thicker rotor wakes downstream of the stator could be moved to any desired circumferential position simply by rotating the IGV row.

The foregoing discussion suggests that it would be useful to distinguish as two quite different phenomena:

1 the rotor wake-downstream stator interaction described by Kerrebrock and Mikolajczak [10], in which accumulation of rotor wake slip flow cannot commence until the rotor wake actually enters the downstream row and the slip flow is intercepted by the stator pressure surface, and
2 the rotor wake—upstream stator wake interaction described by Lockhart and Walker [12], in which accumulation of rotor wake slip flow commences immediately downstream of the rotor trailing edge due to its restriction by interactions with the street of chopped wake segments from the upstream stator.

Similar comments apply to stator-rotor interactions.

Authors' Closure

We are appreciative of the time and effort spent by Dr. Walker in preparing his discussion of our paper.

As mentioned by Schmidt and Okiishi [6], superimposed hot-wire oscilloscope trace data observed by them were in agreement with the

[1] By J. H. Wagner, T. H. Okiishi and G. J. Holbrook, published in the January 1979 issue of the JOURNAL OF ENGINERRING FOR POWER, Vol. 101, No. 1, pp. 42–51.

[2] Mechanical Engineering Department, University of Tasmania, Australia, Mem. ASME.

[3] The reservations expressed about one isolated set of results in [10] in no way invalidate the rest of that paper, which is concerned solely with single-stage machines having an isolated stage of only two blade rows. But it is suggested that the theory presented in [10] will require some modification in the case of an imbedded stage.