DISCUSSION

R. A. Strub[1]

The paper by Dr. Simon and Mr. Bülskämper mentions a working group of the ICAAMC investigating "Workshop Performance Tests on Turbocompressors." This discussion provides some supplementary information about the work of this group.

In 1979 the Process Compressor Subcommittee of the International Compressed Air and Allied Machinery Committee (ICAAMC) decided to investigate the problem of Reynolds number correction between workshop tests and specified process conditions. A working group under my chairmanship was formed to examine the available correction methods and to select the most suitable formulae that could be recommended for incorporation in the appropriate international standards. The following companies agreed to participate on the working group:

- Dresser Industries Inc., Dresser Clark Division
- United Technologies, Elliott Company
- Ingersoll Rand Company
- Transamerica Delaval
- Mannesmann Demag
- MAN-GHH Sterkrade
- Nuovo Pignone S.p.A.
- Sulzer Brothers Limited

The ICAAMC correction equations (given in [35] and included as equations (26), (27), (28), and (29) of the present paper) were the results of many meetings and discussions between the members of this working group. Many different equations were proposed by members of the committee and the best ideas were incorporated into a simple formula that was finally accepted by all participants. Dr. Simon was present at all of our meetings and made many useful contributions.

The paper under discussion, however, gives the misleading impression that the ICAAMC equations were proposed to the ICAAMC working group by Dr. Simon and directly adopted by the working group for comparison purposes with measurements. The fact that the ICAAMC equations can be obtained by simplification of equation (17) of Dr. Simon's paper strengthens this impression, and the reader could be forgiven for believing that they were derived in this way. In fact, the original proposals made by Dr. Simon to this working group were very different from the equations presented in his paper. Discussions within the working group have certainly influenced the development of the ideas and formulae presented by Dr. Simon and Mr. Bulskamper. I therefore regret that the present paper was published by Dr. Simon without informing the working group of the ICAAMC and that no credit has been given to the debt he owes to discussion at its meetings.

The working group of the ICAAMC has compared the predictions of the ICAAMC correction equations with test data supplied by the participating companies. Test data for 31 compressors (17 multistage and 14 single-stage compressors) leading to about 120 test points were submitted by various manufacturers. The deviation of the measured efficiency ratio from the calculated efficiency ratio given by the curve for all of the tests results is plotted in Fig. 21. This figure demonstrates an almost equal distribution of the deviation (scatter) for all sets of data, thus showing that the proposed formula is acceptable for practical purposes. This figure also shows that

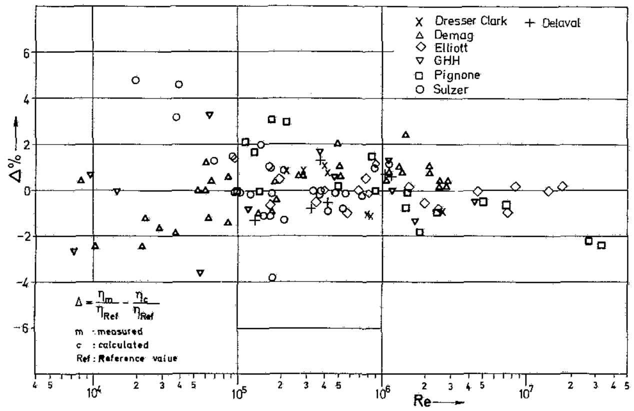

Fig. 21 Comparison between measurements and calculation using the ICAAMC correction equation (equation (26) of Dr. Simon's paper; taken from reference (35).

the scatter for tests at low Reynolds numbers is greater than that at high Reynolds numbers. A large scatter can be expected when performing tests at low pressure or at low peripheral speed, which leads to low driving power.

It was agreed by the participating companies at the beginning of their collaboration on this work that the recommendation of the working group would be jointly presented to ASME and other organizations for incorporation in a revision of existing test codes.

Further References

35 "Influence of the Reynolds Number on the Performance of Centrifugal Compressors," Report of the working group of the Process Compressor Subcommittee of the International Compressed Air and Allied Machinery Committee, finalized Oct. 5, 1982 and issued May 6, 1983, Zurich, Switzerland.

F. J. Wiesner[2]

The authors are to be commended for their valuable addition to the literature on this subject. Although there are a number of questions which should be answered in further clarification of the material presented in this paper, this discusser would appreciate a response to only one in particular. How were the curves labeled "Wiesner" in Figs. 12, 13, 15, 16 and 19 obtained? By starting with the ASME PTC-10 curves shown on these same plots (from which reasonable average values for the "reference" efficiencies can be obtained), this writer has been unable to closely reproduce the trends on these plots which are attributed to the methods of [2]. For the most part, efficiency ratios were arrived at which were higher than shown by the thin solid lines – tending toward the levels of the heavier solid lines which represent the authors' method. With respect to Figs. 12, 13 and 16, in particular, there would appear to be much better agreement between these methods than is implied on the plots.

L. Sapiro[3]

The authors have made an interesting contribution to a subject which still continues to be debatable. However, I do have some reservations regarding their recommendations for incorporating the simplified relationships of this paper into the ASME PTC-10 and ISO TC 118.

My concern stems from the questionable validity of applying conclusions derived from single-stage analytical and actual test models to correct the overall performance of any

[1] Sulzer Escher Wyss Ltd., CH-8023, Zurich, Switzerland, Mem. ASME

[2] Chief Aerodynamicist, United Technologies Elliott, Jeannette, Pa. 15644

[3] Senior Technical Consultant, Solar Turbines, Inc., San Diego, Calif. 92138 Mem. ASME

multistage compressor. As I mentioned during the discussion of Wiesner's paper in the Transactions of the ASME [36], high pressure ratio compressors, especially those handling heavy hydrocarbons, present good examples for the study of this questionable validity. They are critically dependent upon individual stage matching. Thus, a single Reynolds number correction on the overall performance will not be adequate.

A straightforward way to predict performance that accounts for Reynolds number effects as well as Mach number effects consists of correcting each stage individually and working up progressively until arriving at the overall performance. On the other hand, usually it is not possible to arrive at the same overall design performance by applying a single correction to the overall test results.

The formulas for Re number correction, such as the ones presented by this paper and previous ones, are very valuable and should be used for the prediction of performance by applying them to each of the individual stages making up the multistage compressor. However, their suitability for directly correcting the test results is uncertain.

In the majority of cases, test conditions can be selected in such a way as to reduce to a minimum the differences between the friction factor on test and the one at design conditions. This nullifies the need for Reynolds number corrections. However, in those cases where multistage compressors can only be tested at conditions which produce appreciable different Reynolds number effects than at design conditions, the correction should be applied stage by stage.

The correction method for multistage compressors requires a good knowledge of the individual stage characteristics of the configuration being tested, and their matching. Having completed the test, the overall performance has to be compared with its prediction. If there are differences, the individual stage characteristics would have to be adjusted until producing an acceptable agreement with the overall results. Once this laborious task is completed, the individual stages should be corrected for the Re number and used for predicting the overall performance at design conditions.

Regarding the ASME PTC-10 allowable deviations of Reynolds number, it is generally agreed that the present ones are not appropriate and should be revised. The tolerance curves proposed by the authors on Fig. 20 indicate similar trends as the ones of Davis [7] and Wiesner, shown on Fig. 8 of Wiesner's paper. When converting Davis's definition of Re number to the ASME's one, his curves agree very well with Wiesner's, and they are more conservative than the ones of the authors.

Regarding the stages investigated by the authors, it should be noted that they had an inlet flow coefficient ranging from 0.004 to 0.031 (or in the nomenclature of others [37], from 0.005 to 0.039). Those values correspond to a specific speed range of 27 to 76. However, even the stage of the highest value (NS of 76) is smaller than the optimum design value for centrifugal compressors (NS of 90 to 110). Unfortunately, no tests were conducted in the region covering optimum to high specific speed stages to prove the validity of the author's conclusions for any type of single stage.

Also, the authors state that the percentage of losses which are independent of Reynolds number is essentially constant for any value of flow coefficient (specific speed). Thus, stages with extremely narrow impeller and vaneless diffuser passages (NS of 27), where the friction losses predominate, and stages with wide passages of very small radii of curvature (NS of 120), where separation and recirculation losses predominate, would have the same 30 percent of losses independent of Re number (formula (25)). Using the more accurate formula (18), those values would be 19 percent and 29 percent, for NS of 27 and 120, respectively. Those values contradict my studies (formula (5) [37] or Table 1 of [36], and others, which indicated a much higher variation. For the same specific speed extremes my studies would indicate 0 percent and 54 percent, respectively (100 percent and 46 percent due to friction losses, respectively). For the author's stages 3 and 4, they would indicate 5 percent, while for stage 1 they would indicate 41 percent. Other references (Table 1 of [36], indicate extreme values of 15 percent and 57 percent. It is difficult to understand why there is such a small effect of specific speed (inlet flow coefficient) on the percentage losses as indicated by the authors.

References

36 Wiesner, F. J., "A New Appraisal of Reynolds Number Effects on Centrifugal Compressor Performance," ASME Journal of Engineering for Power, Vol. 101, July 1979, pp. 384–396.

37 Sapiro, L., "Preliminary Staging Selection for Gas Turbine Driven Centrifugal Compressor," ASME Paper No. 73-GT-31, Apr. 1973.

Authors' Closure

The authors appreciate the contributions to this discussion and would like to take this opportunity to clear up certain misunderstandings and answer questions concerning the content of the paper.

Dr. Strub's comments only refer to part of our publication—the simplified conversion of measured characteristics from test to specified conditions. In the ICAAMC working group, various conversion methods have been discussed. While the proposals of other members of the working group were derived from the well-known methods of Rütschi [9], Wöhrl [16], and Wiesner [2], we have formulated our own model for the conversion of flow losses (see Fig. 7) which, compared with the others, is the only one which permits physically meaningful conversions free of contradiction. Our model, which provides equation (24) for the conversion of process stage efficiency, has been discussed in detail in the working group. If the conversion of measured characteristics for various roughnesses is waived, the result is the simplified equation (26) which was finally accepted by all participants. The other parts of the conversion method (Fig. 11) were also introduced by us and adopted by the working group without modification, in particular:

- Equation (25), for the Re-independent part of losses
- Equations (27), (28), and (29), for the conversion of head coefficient and work coefficient with a corresponding volume flow coefficient correction
- The transfer of dimensionless characteristics to the new reference points
- The allowable range of application of the conversion method (Fig. 20).

As the members of the ICAAMC working group are well aware, the entire conversion method in all its detail has been determined to a decisive degree by our contributions as is duly documented by the aforementioned facts. We therefore have to reject the objections raised by Dr. Strub. However, we welcome the confirmation of this method which the additionally quoted measurements have provided.

Wiesner's conversion method for efficiency [2] is based on "nominal conditions," which neither coincide with our "reference conditions" nor with the "specified conditions." For this reason, the change in losses must first be converted from Wiesner's nominal conditions to the reference conditions and then from the nominal conditions to the assumed specified conditions. The calculation procedure can be explained in an example with reference to the Wiesner curve shown in Fig. 16. For this compressor, the nominal conditions are as follows: $Re_{NOM} = 52{,}635$; $Ra_{NOM} = 125\ \mu$in. $\hat{=}$ 3.18 μm. For our reference conditions, the following data apply:

$Re_{Ref} = 58{,}200$, $Ra_{Ref} = 62.9\ \mu in. = 1.6\ \mu m$. If one inserts these values in the equation for efficiency conversion (equation (4) in [2]), the result (including a roughness influence $f_{Ref}/f_{NOM} = 0.9767$) is the ratio $(1 - \eta_{Ref})/(1 - \eta_{NOM}) = 0.9704$. For an assumed specified Re number of $Re_{sp} = 10^6$ and the given roughness $Ra_{sp} = 62.9\ \mu in. \hat{=} 1.6\ \mu m$, the result (including a roughness effect of $f_{sp}/f_{NOM} = 0.9373$) is the ratio $(1 - \eta_{sp})/(1 - \eta_{NOM}) = 0.8533$. Division of the changes in loss thus calculated produces $(1 - \eta_{sp})/(1 - \eta_{Ref}) = 0.8793$, which, given the reference efficiency $\eta_{Ref} = 0.5782$, finally results in the efficiency ratio $\eta_{sp}/\eta_{Ref} = 1.0880$. This value is entered in Fig. 16 at $Re_{sp} = 10^6$. The efficiency ratios, plotted in Fig. 12, 13, 15, 16, and 19, can be determined in the same way for any Re number.

In his paper, Wiesner also quotes a range of allowable deviations between the test and the specified Re number which are permitted in the efficiency conversions. The drawn limits are ascertained for a permissible efficiency change of 1 percent. If one applied this principle to our stages 1-5, the result would be a considerable shift of the limits, owing to the flow coefficient. The limits adopted by Davis [6] represent the range in which, according to Davis, the influence of the Re number can be dismissed as negligible. The limits suggested by us imply that for a mean relative roughness, the change in the losses does not exceed 20 percent within the allowable range of application. Within these limits, our method for the conversion of measured characteristics produces for practical purposes sufficient accuracy which corresponds to the usual measurement tolerances.

For stage 5, Sapiro suggests a Re-independent part of losses of $a = 0$ instead of $a = 0.3$ (equation (25)). According to equation (7), the isentropic stage efficiency also includes the leakage losses at the impeller shroud. These losses, which we assume to be independent of the Reynolds number and roughness, constitute for stage 5 alone 22 percent of the total losses. If only the hydraulic losses in this individual stage are considered, the Re-independent part of losses is 19 percent (equation (18)). Apart from low Re-independent losses in the impeller, this percentage is mainly a result of the fact that increasing wall friction and dissipation lead to an increase in the flow angle in the vaneless diffuser and thus to a shorter flow path. An analysis of the diffuser flow shows that for this stage the increase in flow losses only constitutes approximately 60 percent of the increase in the wall friction and dissipation coefficients. For this reason, $a = 0$ cannot be accepted for those stages. In the region of maximum efficiency stages ($0.045 \leq \phi_1 \leq 0.065$), Heidelberg [12] and Weigel [14] have published test results which were taken into account when determining the Re-independent part of losses. Figure 22 shows a comparison of their test results with our calculated curves. The authors themselves describe their efficiency evaluation as unreliable; thus, only the change in the head coefficient was considered. The test results of Weigel correspond well to our calculated curve. The change in head coefficient measured by Heidelberg for a decreasing Re number is greater than the result of the calculation. An increase in the Re-independent part of the losses, as proposed by Sapiro, would further enlarge the deviations between measurement and calculation.

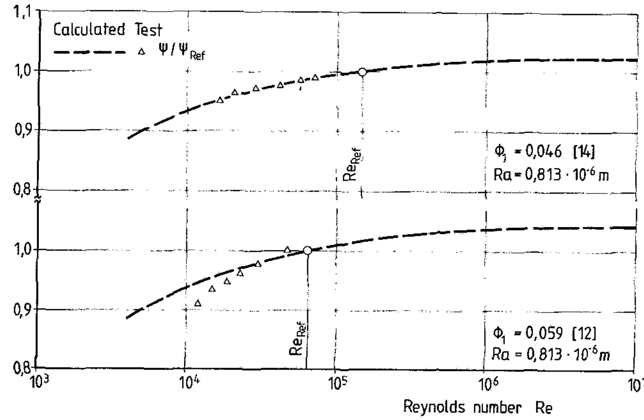

Fig. 22 Comparison of calculated results with test values of Heidelberg [12] and Weigel [14]

The conversion of measured process stage characteristics from test to specified Reynolds numbers can be conducted with the method described in the paper given the prerequisite that all stages are properly matched together. In addition, the acceptance test has to be conducted in strict accordance with the other allowable deviations given in the test codes (e.g., ASME PTC-10 Class II or Class III). Under these conditions the conversion of the test results should produce satisfactory correlation with the results to be expected under the specified conditions (see Figs. 12, 13, 15, 16). If the decisive similarity parameters (e.g., volume ratios, Mach numbers) cannot be maintained during the test, a detailed analysis of the measured process stage performance map is necessary with the aid of the individual stage characteristics, as has also been mentioned by Sapiro. This method has, for example, been applied to the ultra-high-pressure compressor (Fig. 17) because, during the tests, the volume ratios could not be adequately maintained in the inlet pressure range 1 bar $\leq p_1 \leq$ 400 bar.