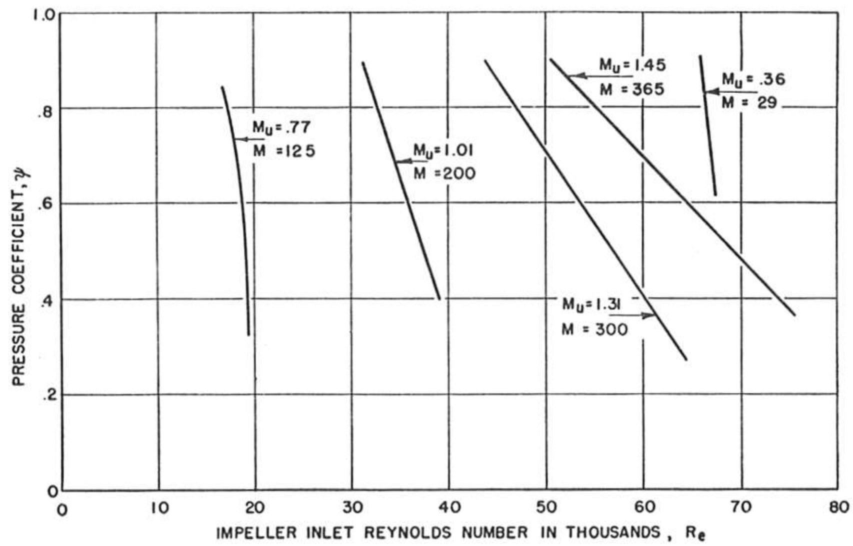

Fig. 12 Pressure Coefficient of Compressor B as a Function of Reynolds Number

Acknowledgment

The author is indebted to the Continental Aviation and Engineering Corporation, Detroit, Mich., for permission to show the data presented in Table 1 and Figs. 3 to 6. The author also wishes to express his appreciation to the Elliott Company, Jeannette, Pa., for permission to present the data in Table 2 and Figs. 7 to 12.

BIBLIOGRAPHY

1 "Applied Hydro- and Aeromechanics," by L. Prandtl and O. G. Tietjens, McGraw-Hill Book Company, Inc., New York, N. Y., 1934.

2 "Fluid Mechanics of Turbo-Machinery," by G. F. Wislicenus, McGraw-Hill Book Company, Inc., New York, N. Y., 1947.

3 "The Theory and Performance of Axial-Flow Fans," by Kurt Keller, McGraw-Hill Book Company, Inc., New York, N. Y., 1937.

4 "Performance Investigations of a Large Centrifugal Compressor From an Experimental Turbojet Engine," by A. Ginsburg, J. W. R. Creagh, and W. K. Ritter, NACA Research Memorandum No. E8H13, 1948.

5 "Centrifugal and Axial-Flow Pumps," by A. I. Stepanoff, John Wiley & Sons, Inc., New York, N. Y., 1948.

6 "An Investigation of Backflow Phenomenon in Centrifugal Compressors," by W. A. Benser and J. J. Moses, NACA Report No. 806, 1945.

7 "The Flow Through Centrifugal Compressors and Pumps," by H. E. Sheets, Trans. ASME, vol. 72, 1950, pp. 1009–1015.

Discussion

J. H. Anderson.[3] This paper provides an excellent analysis of the effect of compressibility on centrifugal-compressor performance. An analysis of this kind is particularly useful in helping the designer to understand the effect of changing volume through the compressor. It is unfortunate, perhaps, that the additional mathematical work and assumptions required for such an analysis tend to confuse the average engineer somewhat, and we find that the simple plot of pressure or head coefficient and efficiency versus inlet volume divided by rpm is the one usually preferred by most engineers. This has the advantage of being simple and straightforward with no assumptions required in the calculations, but it provides very little in the way of a logical mathematical explanation of what happens in the machine.

It has often been the custom to use the tangent of the absolute flow angle at impeller exit as an abscissa instead of the value θ_2 used here. The advantage of using the tangent of the angle lies in the fact that it provides a more accurate idea of diffuser-inlet conditions, particularly when several compressors with different impeller-blade angles are involved. This also becomes more important in a multistage compressor.

If the effect of Mach number on efficiency is to be clearly shown, it would probably be better to use polytropic efficiency instead of isentropic. For example, in Fig. 11 of the paper, the peak polytropic efficiencies would be 0.659, 0.710, and 0.708, as compared with 0.610, 0.675, and 0.685, respectively, for the isentropic efficiencies. These values are not correct because we had neither the isentropic efficiencies for the single stage nor the proper gas constants, but they do illustrate how much effect reheat has, as we go up in Mach number, and they do show that the hydraulic compressor efficiency falls off at Mach numbers well below the design Mach number, as it should.

The concepts of coefficients K_P and K_Q are quite useful and, as pointed out, are particularly so when highly variable Mach numbers are involved, as is the case when the performance of a compressor needs to be estimated for several different gases. There is, however, one slight misstatement which the writer is sure the author did not intend to make. The coefficient K_P is never proportional to the pressure ratio, although it is a function of it as illustrated by the equation.

[3] Ingersoll-Rand Company, Phillipsburg, N. J.

F. Kluge.[4] The dimensionless coefficients φ and ψ, used in practical design frequently, and given in Equations [4] and [5] of the paper, have the practical advantage that they can be determined from readings without too much difficulty with rather good accuracy, while determining the flow coefficient φ_2 in Equation [12] needs more measurements in positions, where accurate measurements are very difficult to make. We know that velocity, pressure, and temperature distribution are changing along the circumference and along the width of an impeller outlet. Therefore, if these readings are not done very accurately, the deviations by mistakes of readings might be bigger than the natural deviations shown in Fig. 5 of the paper.

Considering Fig. 6, it seems to the writer that the coincidence of curves within the range of Mach numbers M_u up to values of 0.99

[4] Engineering Department, Clark Bros. Company, Inc., Olean, N.Y. Mem. ASME.

is a specialty of the tested impeller design, which is not necessarily fulfilled for any design.

As to the dimensionless coefficient K_Q, given by Equation [24], the author refers here to the inlet conditions and not to the outlet conditions of the impeller, evidently for practical reasons, as mentioned.

Since the Mach numbers are defined on the basis of mean velocities, it is not understood how we can get for zero flow ($Q_1 = 0$, $K_Q = 0$) finite values of the Mach numbers M_{w1} and M_{c3}, as shown in Fig. 10, while they have to be zero, since for $Q_1 = 0$ the relative velocity w_1 and the absolute velocity c_3 are zero and, therefore, also the Mach numbers, as defined by Equations [2] and [3].

It might be of value to explain in the nomenclature whether the total head H refers to adiabatic or to polytropic compression.

Author's Closure

It is gratifying to have Mr. Anderson's opinion that the concept of coefficients K_P and K_Q is quite useful for highly variable Mach numbers and different gases. As he points out, it may be better to use polytropic instead of isentropic efficiencies to show the effects of Mach number. However, the test data were taken from the performance of certain compressors for which the isentropic efficiency was determined experimentally. He is correct, and the author regrets the error; the coefficient K_P is not proportional to the pressure ratio but is a function of it as illustrated by Equation [25] of the paper.

The author appreciates and agrees with Dr. Kluge's comments regarding the difficulties of making accurate measurements in the vicinity of the impeller tip.

Considering Fig. 6 of the paper, it is a coincidence of this particular compressor that maximum efficiency and maximum pressure occur at constant values of ϕ_2 up to a Mach number of $M_u = 0.99$ which is above the critical Mach number. However, Figs. 6 and 9 prove that up to the critical Mach number of $M_u = 0.81$ and $M_u = 1.01$, respectively, for the compressors under investigation, maximum efficiency and pressure ratio occur at constant values of ϕ_2.

The author wishes to thank Dr. Kluge for pointing out the finite values of the Mach numbers M_{w1} and M_{w3} at zero capacity in Fig. 10. According to one dimensional theory M_{c1} should have been zero, and some explanation is therefore required. Tests made by the author as well as the data published in references (5) and (6) of the paper, indicate that the flow at the inlet is three-dimensional at zero capacity. This results in an eddy-type flow with inflow at the inner streamline and backflow at the outer streamline of the impeller inlet. While the vector sum of the relative velocity components equals zero, owing to the phase displacement there is a finite scalar value for the relative flow. The mean value M_{c3} is also finite at zero capacity because the mean value of the angle α_3 at the exit of the impeller is zero, and for this condition c_3 approaches u_2. Total head in this paper refers to adiabatic compression.