REPORT 974

THE SUPERSONIC AXIAL-FLOW COMPRESSOR [1]

By Arthur Kantrowitz

SUMMARY

An investigation has been made to explore the possibilities of axial-flow compressors operating with supersonic velocities into the blade rows. Preliminary calculations showed that very high pressure ratios across a stage, together with somewhat increased mass flows, were apparently possible with compressors which decelerated air through the speed of sound in their blading. The first phase of this investigation, which has been reported in NACA ACR L5D20, was the development of efficient supersonic diffusers to decelerate air through the speed of sound. The present report is largely a general discussion of some of the essential aerodynamics of single-stage supersonic axial-flow compressors. In the supersonic flow about isolated bodies, large energy losses usually occur due to wave systems which extend far from the bodies. Supersonic flow entering a cascade is considered and, in this case, the possibility of entirely eliminating this extended wave system is demonstrated; thus, no reason for supersonic compressors to be necessarily inefficient is apparent. The conditions that occur as the flow through the compressor is being started are discussed and a hypothesis as to the type of transonic flow which will be encountered is proposed.

As an approach to the study of supersonic compressors, three possible velocity diagrams are discussed briefly. Because of the encouraging results of this study, an experimental single-stage supersonic compressor has been constructed and tested in Freon-12. In this compressor, air decelerates through the speed of sound in the rotor blading and enters the stators at subsonic speeds. A pressure ratio of about 1.8 at an efficiency of about 80 percent has been obtained.

INTRODUCTION

The occurrence of supersonic velocities in the flow about conventional axial-flow compressor blades results in large energy losses. For this reason, axial-flow compressors are generally designed to operate with relative air velocities into the blade rows low enough to avoid supersonic velocities. This restriction has limited the pressure ratios obtainable with axial-flow compressors to about 1.25 across a stage (rotor and stator) and has meant that many stages are required to produce the pressure ratios needed for gas turbines, turbojet units, and superchargers. It is apparent that great increases in the pressure ratios per stage, with accompanying savings in weight and size, could be obtained if it were possible to design efficient axial-flow compressors with supersonic velocities into the blade rows. The tip speeds required to achieve these supersonic relative velocities and the accompanying high pressure ratios are not in excess of those currently used in impulse-turbine practice (about 1,600 ft/sec). A study is therefore being made at the Langley Aeronautical Laboratory to explore the possibilities of supersonic axial-flow compressors. Introductory general information on the supersonic aerodynamics involved in the present study can be found in reference 1.

Experience with the high wave losses that usually occur in supersonic flows might lead to the supposition that the efficiency of such a compressor would inevitably be low. It must be noted, however, that these high wave losses are usually associated with extended wave patterns. Busemann thus showed and Ferri (reference 2) verified that it was possible to design a biplane in which (in the absence of lift) the waves originating at one airfoil were canceled at the opposite airfoil. In this case, Busemann was able to eliminate the extended wave pattern and, theoretically, all of the wave losses associated with the thickness of isolated supersonic airfoils. This idea can be extended to the case of an infinite cascade and it can be shown that, if the entering Mach number is sufficiently greater than one, the cascade can also produce lift (or turning) without the production of an extended wave pattern, the strong waves being entirely confined to the region between the blades. Thus, there is no a priori reason to suppose that supersonic compressors will necessarily be inefficient, since the usual source of high losses in supersonic flows can be avoided.

Preliminary consideration of the velocity diagrams suitable for use in a supersonic compressor showed that, in order that advantage be taken of the high pressure-rise possibilities, it would be necessary to decelerate the air through the speed of sound in the blading. The first phase of this study (reported in reference 3) was the development of diffusers to decelerate air efficiently through the speed of sound. This study indicated that a normal shock in a diverging passage is necessary for stable flow and methods for minimizing the intensity of the shock were developed. Maximum passage contraction ratio which could be used was found (for the conditions which would probably prevail in a supersonic axial-flow compressor). Diffusers were designed which recovered over 90 percent of the kinetic energy in a supersonic stream up to a Mach number of 1.85. It therefore appears

[1] This report was originally issued as NACA ACR L6D02, April 1946 and until recently has been subject to security regulations.

that supersonic diffusers can be made with efficiencies comparable with those obtained with good subsonic diffusers. Because of the encouraging results of the supersonic-diffuser investigation, a single-stage supersonic axial-flow compressor was designed and built at the Langley Aeronautical Laboratory of the NACA.

The first question to be settled in the study of supersonic compressors is whether the axial velocity should be subsonic or supersonic. A supersonic axial velocity would mean that, as the compressor is started, it would be necessary to form a normal shock in the annulus ahead of the compressor and to move this shock downstream and finally into the rotor. The possibility exists that, in this case, discontinuities in the performance of the compressor would occur when the shock entered the rotor. At the present time, it is not clear how any advantages can be gained by making the axial velocity supersonic (except perhaps in the case of supersonic aircraft in which it might be desirable to avoid the losses inherent in decelerating through the speed of sound in the inlet). The decision was therefore made to study the case of subsonic axial velocities first.

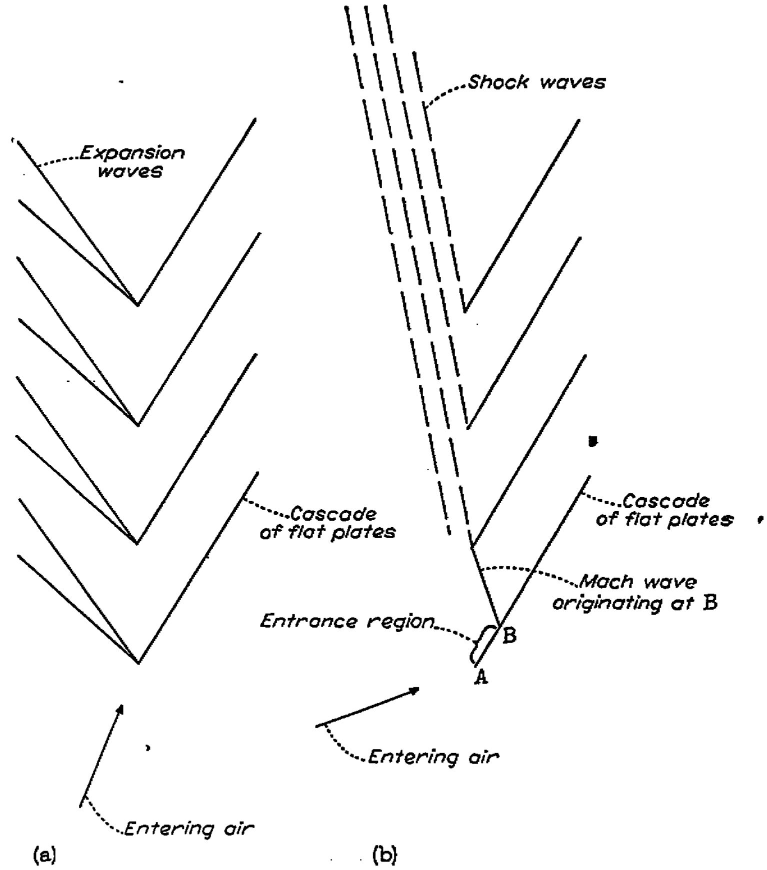
SUPERSONIC CASCADES

ENTRANCE REGION

When the axial component of the velocity entering a cascade is below the local speed of sound, waves started by a portion of each blade can move ahead of the rotor. It will be shown that, in a supersonic compressor, the axial velocity is controlled by these upstream-moving waves. The part of the blades from which upstream-moving waves can originate will be called the "entrance region." Consider Mach waves starting from the rearward side of the blades (that is, rearward as the rotor advances). Schematic illustrations of waves originating in the entrance regions of supersonic cascades are given in figures 1 and 2. Waves originating upstream from the point B in figure 1 can move ahead of the rotor; thus, the region AB of the blading will constitute the entrance region.

The simplest case to be considered is that in which AB is a straight line. In this case, no waves move ahead of the rotor except those given off by the leading edge of each blade (if no strong compression waves are started immediately behind B). The steady flow for this case can be shown to have a velocity vector (in rotor coordinates) parallel to AB.

Consider first the case where the axial velocity is lower than would be required to satisfy this condition (fig. 1 (a)). In this case, an expansion wave corresponding to flow around a corner will be given off from the leading edge A. A similar wave would be given off from the leading edge of each blade. Each of these waves will accelerate the flow in the axial direction until the relative velocity is parallel to AB. Similarly, in the case where the axial velocity is too high (fig. 1 (b)), an infinite series of oblique shock waves will issue from the leading edge of the blades and these waves will reduce the axial velocity until the relative velocity is parallel to AB.



(a) Axial velocity too low. (b) Axial velocity too high.

FIGURE 1.—Schematic illustration showing manner in which volume flow adjusts to cascade geometry for a case with a straight entrance region AB. The only steady flow is that with the entering velocity parallel to AB.

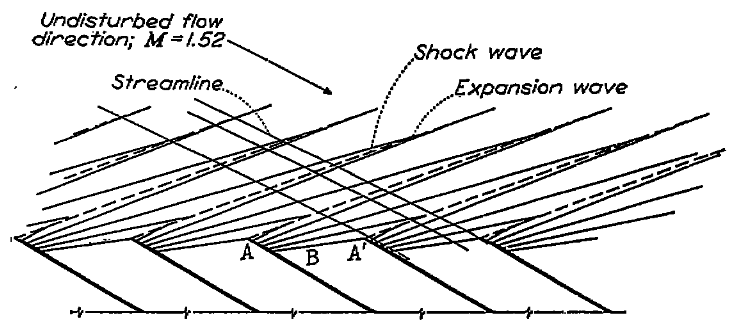


FIGURE 2.—A graphical construction of the extended wave pattern ahead of a supersonic cascade with a curved entrance region AB. The construction starts by the choice of the point B, which determines the conditions far from the cascade. In this construction, the surface AB is a 10° arc of radius equal to 1.38 AA' and the surface at B makes a 30° angle with the stagger line AA'. The energy contained in the pressure waves that move ahead of the blades is enough to heat the incoming air about 1° F (for inlet stagnation temperature 520° F absolute).

Thus, the axial velocity and hence the volume flow at a given rotational speed into the compressor is controlled entirely by the slope of the line AB in the supersonic operating condition. Also, in the steady operating condition, no waves will move ahead of the rotor; the extended wave pattern disappears and all the waves are confined to the passages between the blades. These conclusions do not apply to cascades of a solidity so low that the point B would fall behind the trailing edge of the blades.

It should be pointed out that waves originating from the forward side of the blades (that is, forward as the rotor advances) do not move ahead of the rotor. The shape of the forward side will therefore not affect the entrance velocity so long as the velocity entering the blades is supersonic. (See, however, the subsequent section entitled "Starting Conditions.") If, for example, leading edges of the blades are wedges, the entering velocity will be parallel to the rearward side and an oblique shock will be given off from the forward side.

In practice, the use of a curved entrance region may perhaps be desirable. In this case, waves will be given off, in general, from the whole region AB and will move ahead of the rotor as is illustrated in figure 2. These waves must represent equal expansions and compressions if the condition is to represent a steady state.[1] When the compressions and expansions do not balance each other, the average axial velocity will be changed by the net resultant waves (as in the case considered previously where AB was a straight line) until the waves balance each other. If these waves are to cancel, they must represent equal and opposite deflections of the air stream to a first approximation. The air entering the blades at A must therefore be parallel to the surface at B. The air entering the rotor at other points between the blades, however, and thus the average entering air (that is, along the line AA′) will not, in general, be parallel to the air at A. A graphical method of calculating the wave pattern and the average axial velocity for this case is given in the appendix. The wave pattern found for a particular case is shown schematically in figure 2.

LOSSES DUE TO WAVES MOVING AWAY FROM THE CASCADE

In cases where the entrance region is curved, compression and expansion waves move ahead of the supersonic cascade. If these waves are not reflected by obstacles (such as another cascade immediately upstream), they will be propagated upstream until the compressions and expansions neutralize each other. The energy carried by these waves will be dissipated (largely by shocks) and this dissipation represents a loss in efficiency. Wave losses will also occur when detached "bow" waves are formed ahead of a supersonic cascade (produced by a finite leading-edge thickness or by too large a wedge angle). These bow waves eventually will also be neutralized by expansion waves and this wave pattern will similarly involve an energy loss. Again, in supersonic cascades where the air decelerates through the speed of sound, a normal shock will always be present. There are several circumstances in which this normal shock will not be contained within the passages formed by the blading. The following calculations show that, in this case, large losses appear, as a result of the strong wave pattern propagated away from the rotor. Wave losses of the same kind can occur when air leaves a cascade at supersonic speeds.

[1] It should be noted that a small part of an expansion wave striking a shock is reflected. In these considerations, these reflections are neglected; thus an error is introduced in cases where strong shocks are present.

A first approximation to the energy carried by these waves can be found, once the velocity distribution at the entrance (or exit) of the cascade is known, if the disturbances are assumed to be small and standard relations from the theory of sound are applied. The sound energy transmitted across an element of surface dA in a time dt is $\Delta p \Delta V_{ax}\, dA\, dt$ (reference 4), where Δp is the fluctuation from average pressure and ΔV_{ax} is the fluctuation in the velocity normal to the surface. This expression, which is evaluated in the appendix, is for small disturbances generated by a surface moving with the mean motion of the fluid; however, if the fluid has a mean velocity V_a through the radiating surface as in the case under consideration, the length of the column of sound radiated in a unit time would be reduced by a factor $1-M_{ax}$.

In order to evaluate the effect on efficiency, these losses can be expressed as a temperature rise ΔT by setting the energy dissipation per unit mass flow to equal $c_p \Delta T$, where c_p is the heat capacity at constant pressure. The gas entering the rotor will be heated through the temperature rise ΔT by the waves in the entrance region. The stagnation temperature at the end of the compressor will be raised by ΔT times the ratio of the final stagnation temperature to the temperature before the rotor. This increase in the final stagnation temperature can then be compared with the temperature rise across the compressor to evaluate the effect on the efficiency. It is shown in the appendix that

$$\Delta T = \frac{\overline{\Delta V_{ax}^2}}{c_p M_{ax} \cos \theta} (1 - M_{ax})$$

where M_{ax} is the axial Mach number, θ is an average angle between the waves leaving the cascade and the stagger line (defined in fig. 3), and $\overline{\Delta V_{ax}^2}$ is the mean-square fluctuation in axial velocity at the entrance to the cascade. The wave losses are then roughly proportional to the square of the departure of the entrance region from a straight line. Evaluation of these energy losses for the wave system shown in figure 2 may be of interest. By graphical integration, it is found that $\overline{\Delta V_{ax}^2}$ is 9,100 (ft/sec)^2 and, with the shock angle (23°) used as θ, there is obtained $\Delta T = 1°$ F for air. These

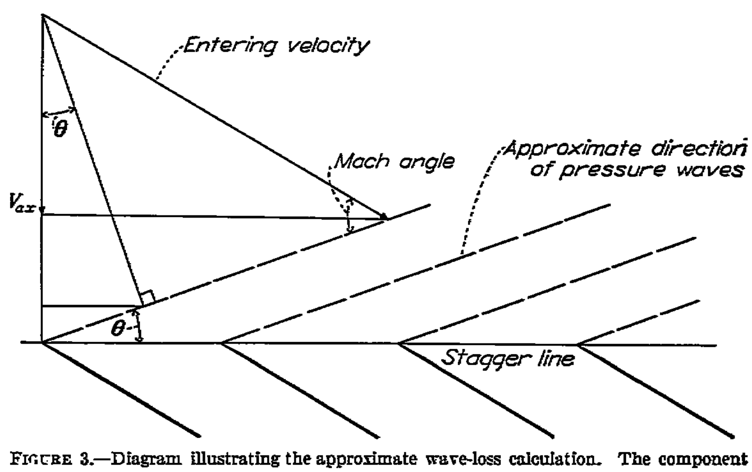


FIGURE 3.—Diagram illustrating the approximate wave-loss calculation. The component of the entering velocity normal to the wave direction is approximately the local speed of sound. All velocity changes are normal to the waves.

losses will not be very large so long as all the waves are attached to the blades. For cases in which detached bow waves are formed or in which a normal shock is not contained within the blade passages, the root-mean-square variation in axial velocity can increase to several hundred feet per second and the resultant wave losses are prohibitive.

The losses considered in this calculation are only those upstream of the line at which the mean-square variation in axial velocity is known. For example, if bow waves are present, the losses due to the portions of these waves behind the leading edge of the cascade are not considered.

In the event that it is necessary to calculate losses due to a system of strong waves more accurately than this first approximation, the mean-square variation in axial velocity can be calculated some distance ahead of the cascade where the strength of the waves will be somewhat reduced. If this loss is then added to the losses occurring downstream from this line (which can be found directly from the known shock strengths), a more accurate estimate of the energy losses can be obtained.

STARTING CONDITIONS

It is important in planning any type of supersonic flow, and especially flows involving deceleration, to examine the mechanism by which the flow is to be set up. For example, for the supersonic diffusers investigated in reference 3, the maximum amount of contraction of the supersonic passage that could be used was found to be determined by the starting conditions. In this case, a normal shock formed ahead of the diffuser and the critical condition was with the shock at the entrance. It was necessary to ensure that, in this condition, the mass flow passing through the shock could also pass through the throat of the diffuser. In the case of a supersonic compressor, the starting mechanism is much more complicated and is not at present fully understood. Speculation on this subject, however, may be of use in providing guidance for further experimental work.

The axial velocity entering the compressor will be assumed to be always subsonic. As the rotational speed of a compressor is increased, the relative velocity into the blading will reach and exceed the local speed of sound. It is clearly not possible to have a uniform stream at the local speed of sound entering a cascade of blades of finite thickness since in this condition the mass flow per unit area is already a maximum. Consider, however, a system of detached bow waves formed ahead of each blade and compensated by expansions starting from the blade surfaces, such as those shown schematically in figure 4. Now, along the stagger line AA′ the average mass flow per unit area is everywhere less than maximum and it should be possible thus to allow for the space taken up by the blades. Such a bow-wave system has been observed to form in the flow about a stationary subsonic cascade even before the speed of sound is reached ahead of the cascade. The flow about an isolated body appears to be fundamentally different from the flow into a cascade in that the bow waves will not move indefinitely far ahead of the cascade when the entering velocity is close to sonic.

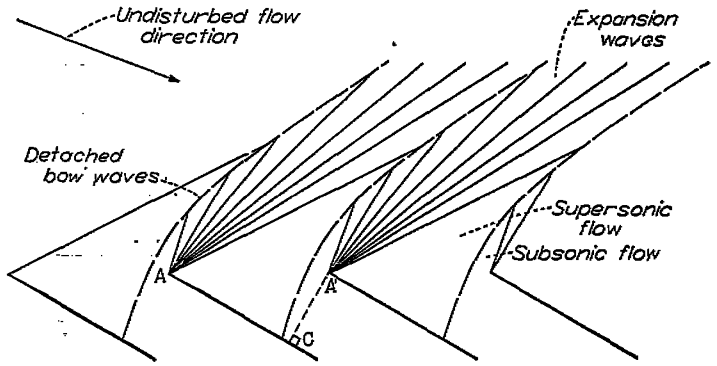


FIGURE 4.—Hypothetical wave pattern for cascades in the transonic region. As the entering Mach number is increased, the detached waves move closer to the blade leading edges; thus, the energy radiated from the cascades is reduced. A mechanism of this kind permits the air to pass through the cascade even when the undisturbed velocity is close to sonic. The velocity crossing the line AA′ is partly supersonic and partly subsonic; thus, the average mass flow per unit area can increase to allow for the thickness of the blades.

A bow-wave system would have two effects on the performance of the compressor: first, the wave losses considered previously would occur and, hence, the efficiency would be reduced; and second, the axial velocity and the mass flow would be lower than would be expected for a supersonic operating condition.

As the rotational speed of the compressor—and consequently the relative velocity entering the blades—is increased, a weaker wave system (bow waves closer to the blade leading edges) is required to permit the passage of the air through the blades. Thus, as the rotational speed was increased, it would be expected that a favorable effect on the efficiency would be obtained and that the relative velocity at entrance to the blading would become more closely parallel to the line AB. (See fig. 1.) If this starting mechanism is correct, it would also be expected that the starting of a supersonic compressor would not occur discontinuously but that gradual changes would occur as the bow waves approached the blade leading edges. Results on the experimental supersonic compressor described subsequently tend to support these three conclusions based on the bow-wave hypothesis.

In order for the bow waves to become attached, the air crossing the line A′C (fig. 4) must be able to pass through the narrowest section of the blading. When the bow waves are nearly attached, the conditions just ahead of the downstream leg of a bow wave will be close to the operating supersonic conditions. The mass flow in this condition can be approximately calculated from this fact. Attachment can take place only if this mass flow, after experiencing the losses in the downstream leg of the bow wave, can pass through the narrowest section of the blading. The losses in the downstream leg of the bow wave will be of the order of magnitude of normal shock losses at the entrance Mach number. Perhaps, therefore, a good first guess as to the ratio of A′C to the minimum passage area that could be used in supersonic compressor blading would be the contraction ratio found previously for supersonic diffusers (fig. 3 of reference 3). This hypothesis is at present hardly more than a speculation and much more work will have to be done before the starting mechanism is fully understood.

One other necessary condition is that the wedge angle be made small enough to allow attachment of the bow wave (fig. 2). Note that the air must turn through an angle equal to the total wedge angle in the case of a straight entrance region.

Present indications are that compressors operating with detached bow waves will not be very efficient. In order that the bow waves become attached to blades with a practicable thickness near their leading edges, the relative Mach number must be considerably above 1; thus, compressors operating with relative Mach numbers near unity will probably not be of much practical interest.

THE DESIGN OF SUPERSONIC COMPRESSORS

In order to stimulate thinking about supersonic compressors, a few representative single-stage designs are discussed briefly.

Preliminary remarks.—In general, the energy input per unit mass of an axial-flow compressor stage can be shown from simple momentum considerations to be the product of the blade peripheral speed V_B and the change in tangential velocity ΔV_T. All this energy appears as heat producing a stagnation temperature rise $\Delta T=\frac{V_B\Delta V_T}{c_p}$ where c_p is the heat capacity at constant pressure in mechanical units (6,006 ft-lb/slug/°F for room-temperature air).

If the efficiency of the compressor is 100 percent, the pressure ratio per stage is

$$\text{P.R.}=\left(\frac{T_o+\Delta T}{T_o}\right)^{c_p/R}$$

$$=\left(1+\frac{V_B\Delta V_T}{c_pT_o}\right)^{c_p/R}$$

where T_o is the initial stagnation temperature and R is the gas constant for air. If the adiabatic efficiency is η,

$$\text{P.R.}=\left(1+\eta\frac{V_B\Delta V_T}{c_pT_o}\right)^{c_p/R}$$

It was pointed out in the section entitled "Supersonic Cascades" that, in order to permit the use of blades with sufficient thickness to be structurally practical, the Mach numbers used must be considerably above 1. For this reason and because the pressure ratios obtainable increase quite rapidly with the blade speed available, the blade speed in these typical designs is assumed to be 1,600 feet per second at the tips. Since tip speeds of this order of magnitude have been used in practical turbines, which have been run at high temperatures, it seems quite likely that the construction of an axial-flow compressor to run at this speed will be practical mechanically, at least from the centrifugal-force considerations.

The inherent advantages of supersonic compressors are: first, high pressure ratio per stage, and second, high mass flow. The chief apparent disadvantages are the mechanical difficulties usually associated with high-speed machines. It seems, therefore, that the first applications of supersonic compressors will be to high power machines, where compactness and light weight are important. For these applications, it will be desirable to make the mass flow large. Maximum mass flow is obtained when the velocity at the entrance to the first rotor is axial and at the local speed of sound. The maximum mass flow is thus obtained without the use of entrance vanes. Other things being equal, however, it is possible to obtain an increased pressure rise by the use of entrance vanes which turn the air against the direction of rotation. Entrance vanes can certainly be of use in equalizing the energy input at the root and tip of the blades. The design of entrance vanes does not now seem to present any new problems and, therefore, these vanes have not been considered in the designs presented herein.

For the cases discussed, only the tip flow will be considered and it is not yet known how far the blades can be continued toward the axis (that is, how large the annular air passage can be made) before serious structural or aerodynamic difficulties intervene. Thus, it is not yet clear how large the mass flow of supersonic compressors of a given diameter can be made. However, it seems quite certain at present that the mass flow can be made larger than that of any existing subsonic axial-flow compressors. In the designs considered, the axial velocity has been made subsonic. In the discussion that follows, the passage-area changes can be assumed to be provided, where necessary, by varying the annulus areas from the front to the rear of the blades.

Design 1; supersonic relative velocity entering and leaving rotor and subsonic stator.—The first design considered (design 1) avoids the necessity of decelerating air through the speed of sound in any part of the compressor. The pressure ratio that can be realized in this type of compressor is seriously limited. In order to keep the mass flow through the compressor high, the entering axial Mach number must be fairly high—say, of the order of 0.8. The static pressure ahead of the rotor will therefore be 0.66 times the initial total pressure for axial entry. If the initial-air stagnation temperature is 520° F absolute and the tip blade speed is 1,600 feet per second, the Mach number entering the rotor blades will be about 1.72. If the maximum pressure ratio is to be obtained, it will be clearly desirable to obtain the highest possible static-pressure ratio in the rotor since the static-pressure ratio in the stator is definitely limited. In order to increase the static pressure in a supersonic passage without decelerating through the speed of sound, the passage must be contracted. The maximum contraction ratio found from figure 3 of reference 3 that permits the air following a normal shock at a Mach number of 1.72 to pass through the passage is 1.15. In the supersonic operating condition, the Mach number after a contraction of this magnitude would be 1.51. If this value, then, is the Mach number leaving the rotor, the static-pressure ratio across the rotor would be about 1.37. (See reference 1.)

A second limitation for this design is due to the subsonic cascade behind the rotor. Subsonic cascades have been designed for high efficiency up to a Mach number of 0.8.

At this Mach number, the total pressure entering the stator is 1/0.66 times the static pressure p entering the stator. In the absence of any losses, the total-pressure ratio across the single-stage compressor would be $0.66\times 1.37\times\frac{1}{0.66}=1.37$. A velocity diagram for this case is given in figure 5. This calculation indicates that the maximum pressure ratio which can be obtained from supersonic compressors which do not anywhere decelerate the air through the speed of sound is limited to the same order of magnitude as that obtainable with subsonic axial-flow compressors and that the chief advantage of such a compressor would be in its high mass-flow possibilities.

Design 2; deceleration through the speed of sound in rotor and subsonic stator.—The design of blading to decelerate air through the speed of sound has not been given sufficient study to make clear what can or cannot be done. An indication of the type of rotor flow that will be desirable for maximum pressure rise can be obtained from consideration of a compressor in which the air enters the stator with the maximum tangential velocity for which efficient subsonic stators can be designed. For a stator Mach number of 0.8 and a stagger angle of 60°, the pressure ratio across the compressor can be calculated.

Assume a blade tip speed of 1,600 feet per second and an inlet stagnation temperature of 520° F absolute. From the stator design conditions, the ratio of the tangential velocity entering the stator (V_T) to the local speed of sound can be found. This ratio leads to a known value $V_T/\sqrt{T_2}$ where T_2 is the stagnation temperature behind the compressor. It was shown previously that

$$T_2=T_o+\frac{V_B\Delta V_T}{c_p}$$

where, in this case, ΔV_T can be replaced by V_T. This equation can be solved for the conditions given and yields $T_2=754°$ F absolute. The pressure ratio for 100 percent efficiency is 3.67 or for 80 percent efficiency 2.93. The pressure rise determined by these limiting conditions is unaffected by the axial velocity entering the rotor. This axial velocity will therefore be chosen as 0.8 of the local speed of sound, which will yield a high mass flow. The states of the air at various points now can be readily calculated, if losses are neglected, and are given on the velocity diagram (fig. 6).

This calculation indicates that a compressor of this type would have considerable advantages over subsonic axial-flow compressors. The most significant problem presented is in the design of rotor blading to decelerate air from a Mach number of 1.72 to a Mach number of 0.68 and simultaneously to turn the flow through an angle of 7.36°. The preliminary work on supersonic diffusers reported in reference 3 was undertaken to study this problem. It was found possible to design supersonic diffusers with efficiency comparable with that of good subsonic diffusers. Machines decelerating air through the speed of sound were therefore considered to have considerable promise and construction of an experimental supersonic compressor of this type was decided upon. This compressor and preliminary test results are described briefly later in the present report.

The results of reference 3 clearly indicate that the deceleration through the speed of sound in the rotor passages must be accomplished by a normal shock. The efficiency of this compressor then depends upon how much the total-head and separation losses accompanying this normal shock can be reduced. These losses can be reduced if the Mach number immediately ahead of the normal shock can be lowered. The Mach number ahead of the normal shock can be lowered by starting compression waves as soon as possible behind the entrance region. The tests showed that starting compression waves by the inclusion of a concave region on the rearward side of the blades—that is, rearward as the compressor rotates—immediately behind the entrance region considerably improved the efficiency of the test compressor. (See fig. 7.)

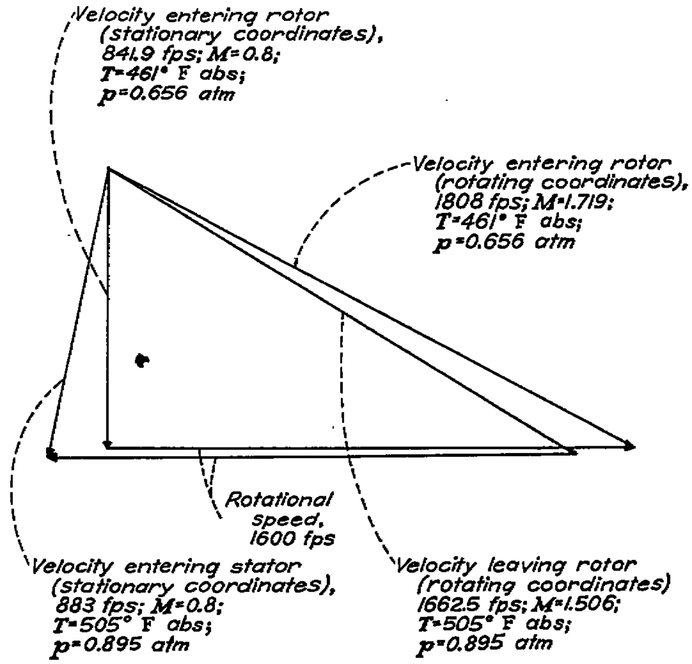


FIGURE 5.—Velocity diagram for supersonic compressor 1. In this design, air enters and leaves the rotor at supersonic speed and the Mach number entering the stator is limited to 0.8. Starting considerations limit greatly the amount of deceleration which can be used in the rotor and the 0.8 stator Mach number condition severely limits the amount of turning in the rotor. Thus, the energy input and the pressure rise (even for the isentropic flow assumed here) are small.

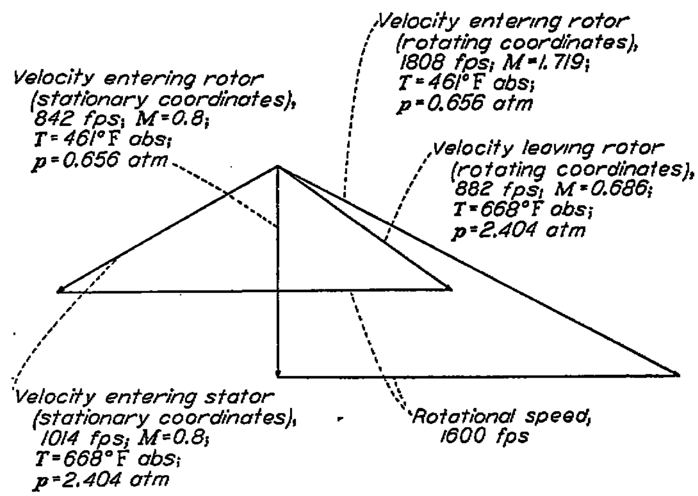


FIGURE 6.—Velocity diagram for supersonic compressor design 2. In this design, the air enters the rotor at supersonic speed and the Mach number entering the stator is limited to 0.8. The pressure ratio obtainable by a machine of this type is restricted to about 3 by this limitation. This velocity diagram assumes isentropic flow throughout and for this case the pressure ratio would be about 3.67.

The results of reference 3 showed that very slow divergence of the subsonic passage was necessary to obtain high efficiency in the supersonic diffusers tested. A machine of this type would consequently be expected to require passages of considerable length—compared to the distance between blades—in order that the passage divergence might be slow enough to prevent serious separation losses.

In the tests of reference 3, it was found that the position of the normal shock in the diverging portion of the diffuser was stable and was controlled by the back pressure at the subsonic end of the diffuser. In the supersonic compressor (design 2), the position of the normal shocks in the blading similarly would be expected to be controlled by the back pressure so long as the shocks were located in portions of the blading where the passage area was diverging downstream. The losses in the normal shocks would thus be reduced as the back pressure on the compressor was increased. The maximum efficiency would be expected to occur just before the normal shock was forced out ahead of the compressor blading. When the normal shock is forced out ahead of the blading, a strong extended wave system is set up ahead of the compressor and large losses result. This condition will most likely produce a result similar to stalling in subsonic compressors.

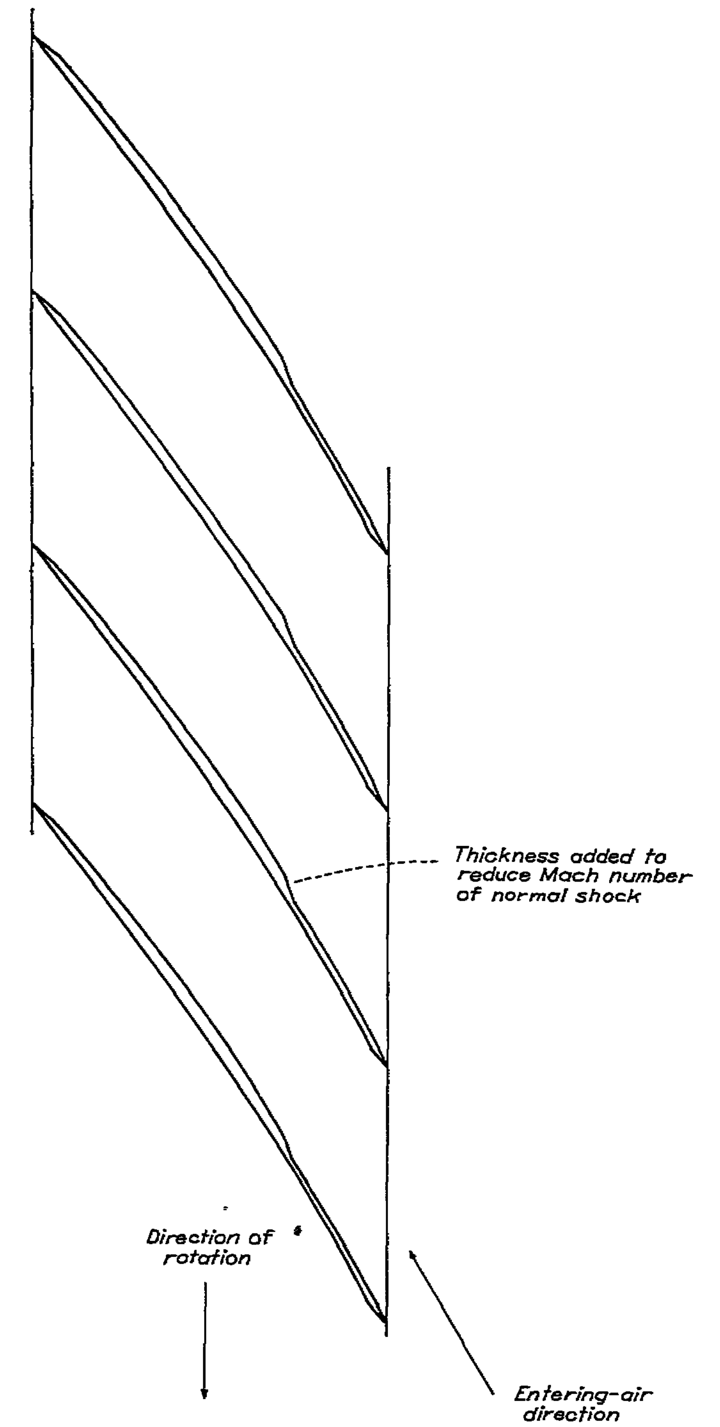


FIGURE 7.—Cross section at pitch of blades of experimental supersonic compressor. In order to prevent flow separation, the annulus in which the blades run was made to contract downstream.

Design 3; deceleration through the speed of sound in the stator.—The third design to be considered (design 3) provides deceleration through the speed of sound in the stator. This type has been given very little study and not much can be said about it at present. (No velocity diagram is presented for this case.) If deceleration through the speed of sound is permitted in the stator, there are no limitations on the usable velocity diagrams that are apparent at present. The pressure ratio obtainable will, as far as can now be foreseen, depend only on the tip speed available. Of course, in this design—as in design 2—it will be necessary to design passages that decelerate or turn the air flow without serious separation losses. A difficulty that this design introduces is that the solidity of the blading used will have to be considerably higher than the solidities ordinarily employed in axial-flow compressors. For example, if a rotor blade design that turns the air back to the axial direction relative to the rotor can be used, the tangential velocity imparted to the air by the rotor would then be equal to the tip speed. If the initial stagnation temperature and blade speed are again 520° F absolute and 1,600 feet per second, the final temperature coming from such a compressor would be 946° F absolute. Assume an efficiency of 80 percent; the pressure ratio would then be 5.85. Such a compressor will probably present new starting problems and these, as well as efficiency considerations, will determine the velocity diagram used. This type of compressor will need a great deal of study and it is hoped that more attention can be given to this design in the future.

PRELIMINARY RESULTS ON THE EXPERIMENTAL SUPERSONIC COMPRESSOR

An experimental supersonic compressor, which is designed to decelerate the air through the speed of sound in the rotor and to operate with subsonic stators, has been constructed and preliminary tests have been run. The compressor is designed to operate at a tip speed of 1,600 feet per second in air, with an entering stagnation temperature of 520° F absolute. The tests were actually run at comparable Mach numbers in Freon–12. (As far as is known, Freon–12 can be used to simulate the flow of air, at least qualitatively.) This gas has a velocity of sound about half that of air and many of the structural problems could therefore be avoided in these preliminary tests.

The compressor was originally intended to have approximately the velocity diagram of design 2 discussed previously. The first tests, however, disclosed that the passages formed by the rotor blading diverged too rapidly in the subsonic region and serious separation losses resulted. It was necessary to reduce the rate of this divergence and the annulus was therefore made to converge downstream. This change reduced the passage area at the exit of the rotor blading and the Mach number leaving the rotor blading was therefore about 0.97 instead of 0.68, as intended in the original design. This modification decreased materially the pressure ratio obtainable from this compressor.

In order to minimize the starting difficulties, the blades, which are shown schematically in figure 7, were made very thin (thickness, about 2 percent chord). Since the tests have been run, it seems that these difficulties were overemphasized and that thicker blading can be used. In these blades the entrance region was nearly straight and the necessary thickness (0.015 in. near the leading edge) was provided by a 10° wedge. The blade length (2 in.) was made one-quarter of the tip radius so that the theoretical volume flow through this machine would be about as large as any previously obtained in axial-flow compressors of similar diameter. As remarked previously, it is not yet known how great the blade length can be made compared to the tip radius, but no great difficulties appear to have been introduced by the blade length selected. Entrance vanes ahead of the rotor were used to help equalize the energy input supplied by the root and tip of the blades.

Tests were made by surveying total head, static pressure, and flow direction in front of and behind the rotor. The test results do not include any losses that might occur in the stator behind the rotor. The results obtained thus far indicate that:

(1) No discontinuity occurs in the performance of this compressor in passing through the transonic region with the blading used. All indications are that the transition from a subsonic type of flow to a supersonic type of flow occurs smoothly. The data in the transonic region provide some support for the bow-wave starting-mechanism hypothesis discussed previously.

(2) Pressure ratios of the order of 1.8 with efficiencies of about 80 percent are obtainable with this compressor in its present form. It should be remarked that considerable warping occurred during the fabrication of the thin blading of this compressor and that this warping resulted in large inaccuracies of construction. Plans are consequently being made in the Fluid and Gas Dynamics Section of the Langley Laboratory to construct a compressor with thicker blading and with a higher solidity in an attempt to realize the performance predicted in the discussion of design 2.

(3) The volume flow entering the rotor was close to the theoretical value and was virtually independent of the back pressure on the compressor in the supersonic operating condition in agreement with the theory of the entrance region of supersonic cascades discussed previously.

(4) The efficiency of this compressor could be increased by starting compression waves on the rearward sides of the blades (that is, rearward as the compressor rotates) just behind the entrance region. Compression waves were started by adding thickness to the rearward side of the blades, as shown in figure 7. Improvements in efficiency of several percent were made in this way.

CONCLUSIONS

Supersonic axial-flow compressors are capable of much higher pressure rises per stage and probably somewhat higher mass flows than are possible with subsonic compressors. The extended wave system, which is usually responsible for the large losses in the supersonic flow about isolated bodies, can be eliminated in the case of a supersonic cascade and thus there is no apparent reason to indicate that supersonic cascades would necessarily be inefficient.

A hypothesis as to the type of transonic flow that will occur as the compressor is brought up to operating speed is proposed. It appears from this hypothesis that the starting of supersonic compressors designed for subsonic axial velocity will not involve any discontinuities.

The volume flow entering a supersonic compressor (in its operating condition) is independent of the back pressure.

A single-stage experimental supersonic compressor that was tested in Freon–12 showed a pressure ratio of about 1.8 at an efficiency of about 80 percent. No discontinuities in performance while the compressor was being accelerated could be observed. A theoretical method of calculating the operating volume flow was presented and agreed well with experiment.

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National Advisory Committee for Aeronautics,
Langley Field, Va., *March 8, 1946.*

APPENDIX

WAVES CREATED BY A CURVED ENTRANCE REGION

Graphical construction of the extended wave system.—All waves involved in the flow ahead of the cascade originate on the rearward surface of the blades and are therefore of the same family. All the oblique shocks present will be assumed to be of sufficiently low intensity to permit the flow to be considered isentropic to a first approximation. These facts allow the flow to be constructed graphically as follows: First, assume a point B (fig. 2) on the rearward side of the blade to be the end of the entrance region. The choice of this point B will be shown to determine the rotational speed (or more accurately, the ratio of the rotational speed to the speed of sound). Draw the line BA′. This Mach wave is the last in the entrance region. The Mach angle—and hence, the local Mach number—can now be determined by measuring the angle between BA′ and the blade surface at B. If a real flow is to be represented, this angle must be less than 90°. The Mach lines originating at points between B and A can now be drawn. (The velocity along AB is assumed to be everywhere supersonic.) It has been shown that, for steady flow, the air entering the rotor at A must be parallel to the surface at B. The wave (in the case illustrated, an oblique shock) originating at A can then be drawn. If this process is repeated for successive blades, a complete flow pattern can be found. The compression and expansion waves interact and eventually cancel each other some distance ahead of the rotor. From the construction given in figure 2, the Mach number and direction of the air far from the cascade can be found in rotating coordinates. The direction of the air far from the cascade in stationary coordinates will be known from the known setting of stator blades ahead of the rotor. (In the absence of any stator blades, it would be assumed that the undisturbed velocity was axial.) It is now possible to complete the velocity triangle and thus to find the rotational velocity and the undisturbed axial velocity and volume flow.

Losses due to an extended wave system.—The losses involved in the waves that move ahead of or behind the rotor can be calculated once the wave pattern is known. A first approximation to the energy carried by these waves can be found by assuming them to be small disturbances and applying well-known relations from the theory of sound (reference 4). The sound power transmitted across a surface A (chosen perpendicular to the axis) is given by

$$\int \Delta p \Delta V_{ax} \, dA$$

where Δp is the fluctuation in pressure and ΔV_{ax} is the fluctuation in velocity normal to the surface. This expression is for small disturbances generated by a surface moving with the mean motion of the fluid; however, if the fluid has a mean velocity V_a through the radiating surface as in the case under consideration, the length of the column of sound radiated in a unit time would be reduced by a factor $1-M_{ax}$. If the disturbance is produced by pressure waves—such as those shown in figure 2—all velocity increments are normal to the waves, there being no velocity change in the plane of the waves. Since the component of velocity normal to the wave front is approximately the local velocity of sound a, the pressure jump Δp across a wave front will be

$$\Delta p = \rho a \Delta V_a (1 - M_{ax})$$

where ρ is the ambient density and ΔV_a is the change in the velocity normal to the wave front in crossing the wave. From figure 3, it can be seen that ΔV_{ax} in this case is given by $\Delta V_{ax} = \Delta V_a \cos\theta$; therefore,

$$\Delta p = \rho a \frac{\Delta V_{ax}}{\cos\theta}$$

The energy dissipated per unit mass entering the compressor is then

$$\frac{1}{A \rho_o a_o M_{ax}} \int \rho a \frac{\Delta V_{ax}^2}{\cos\theta} \, dA$$

where ρ_o and a_o are density and sonic velocity, respectively, in the undisturbed stream and A is the annulus area. Since the analysis is restricted to small disturbances, it will be permissible to assume constant values of density, sound velocities, and θ; these quantities thus can be taken out of the integral. Equating this power to a heat input $c_p \Delta T$ gives

$$\Delta T = \frac{\overline{\Delta V_{ax}^2}}{M_{ax} \cos\theta \, c_p}$$

where

$$\overline{\Delta V_{ax}^2} = \frac{1}{A} \int \Delta V_{ax}^2 \, dA$$

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