High Pressure Ratio Axial Flow Supersonic Compressor

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The present invention relates to axial flow fluid compressors and, more particularly, it is concerned with the construction of an axial flow compressor capable of producing a high pressure ratio from a single axial flow stage, when operating under relative supersonic compressor inlet conditions.

With the advent of jet aircraft and other types of aircraft capable of operating at or above the speed of sound, the problem of air compression to provide air for the burning of fuel, increased materially. It was discovered early in testing operations that conventional axial flow compressors designed for operation in the subsonic range (velocities below Mach 1) are entirely unsatisfactory for compression of fluids entering the compressor at velocities at or above Mach 1. Accordingly, test procedures and structures were developed for determining the optimum compressor structure for operation at supersonic velocities. As a result of this comprehensive design construction and testing program supersonic axial flow compression of air or the like has been proved practical and efficient. In a prior supersonic compressor developed in this program efficient compression of air entering the system at velocities above Mach 1 was found practical for providing a pressure ratio of approximately 2.5 to 1 for a single stage. While this prior compressor was found to be very satisfactory relative to flow efficiency and accordingly was an important advance in the field, the pressure ratio for efficient operation was limited to the above noted values of approximately 2.5 for a single stage. While this is an unusually high pressure boost per stage for supersonic compressors heretofore known in the art, it is very desirable to provide, if possible, a compressor capable of substantially higher ratio, preferably in the range of 4 to 6 in a single compression stage.

Prior to the instant application no such high pressure ratio single stage compressor of a supersonic type was known. In fact, among the leading researchers and writers on the subject substantial doubt existed as to whether or not such a compressor could be constructed. The major difficulty facing those skilled in the art in obtaining high pressure ratios at supersonic speeds was the necessity of providing a high degree of fluid turn during passage through the rotor efficiently. To our knowledge, no prior art has successfully overcome the difficulties involved in turning fluid at supersonic velocities without extreme flow separation in such a manner as to provide a compressor capable of providing extremely high pressure ratios in a single rotor-stator stage. The structure of the present invention satisfactorily provides such a compressor through the use of blading, either in the rotor and stator or in the stator only which is capable of accepting fluid flow above the speed of sound, decreasing the speed or the velocity of flow down through the speed of sound and subsequently turning the fluid flow a substantial number of degrees subsonically and with diffusion.

Compressors for supersonic operation having both supersonic rotor and supersonic stator, requiring a large angle of fluid flow turn in the rotor, have previously been considered impractical because of the high turning angle required and the impossibility, then believed, of providing for such high angles of turn in the range of supersonic operation. Through experimentation, we have found that very high turning angles can be incorporated in supersonic blading where such blading provides a normal shock, positioned where the fluid flow passes downwardly through the velocity of sound, with subsequent diffusion, if the subsequent diffusion is carefully controlled by limiting the expansion of the channel area downstream of the throat or point of normal shock location to a small percentage while the flow is being turned. By further providing high blade solidity in order to provide adequate support for the flow through the high turning required and by using sharp blade leading edges having a small blade included angle in order to prevent the build-up of strong oblique shocks providing, instead, a series of low loss weak oblique shocks efficiently diffused in the flow, the outer and inner passage contours of both the rotor and the stator are controlled, as will hereinafter more fully be developed, to provide radial outward flow turn to establish radial equilibrium and prevent a substantial pressure gradient, occurring between the inner and outer confines of the fluid flow channel.

By providing blading capable of satisfactorily turning fluid flow through large angles of turn and at the same time diffusing through the speed of sound, it has been possible to provide two types of single stage compressors capable of producing high pressure ratios heretofore considered impractical by prior art engineers. These comprise first, a compressor of the impulse rotor type having satisfactory overall efficiency has been constructed. The compressor provides a high degree of flow turn in the compressor rotor without any velocity change and hence without a normal shock in the rotor passageway. In view of this condition the fluid leaves the impulse rotor at a velocity not only supersonic relative to the stator but supersonic relative to the rotor blade and accordingly substantially higher velocity relative to the stator than in a shock-in-rotor compressor design. The flow leaving the impulse type rotor must be turned to the axial direction and reduced in velocity, which functions are achieved through the use of the blading above described wherein diffusion is very carefully controlled and the passageway area expansion is maintained at a small figure. Preferably, in this class of compressor, the fluid flow passes through the speed of sound, and hence the normal shock position is located, in the stator blading to provide optimum performance. However, the compressor is operable even though the fluid flow is diffused through the speed of sound downstream of the stator blades after efficient high velocity fluid turning in the stator blades.

The second, as a result of the development of the above described blading, is an axial flow compressor having a supersonic rotor wherein the normal shock is positioned in the rotor and a subsonic relative fluid flow velocity is provided at the exit of the rotor blades, and having a supersonic stator, likewise preferably having the normal shock position therein. Both the rotor and stator have a high degree of flow turn providing unusually high overall pressure ratio.

It is therefore an object of the present invention to provide a supersonic compressor having an unusually high overall pressure ratio, in the range of 4 or more per stage.

Another object of the present invention is to provide a supersonic compressor providing a high angle of fluid turn of fluid having a velocity which is at some point in the system above the speed of sound.
Still a further object of the present invention is to provide efficient compressor blading capable of diffusing supersonic air flow through the speed of sound and simultaneously turning the flow through a large angle without appreciable flow separation and turbulence.

A feature of the invention is the construction of supersonic blading capable of diffusing high fluid flow through the speed of sound by means of a series of oblique shock waves followed by a normal shock and subsequent high angle flow turn with a nominal controlled amount of channel area expansion.

Still a further object of the present invention is to provide a novel impulsive type supersonic compressor wherein supersonic fluid flow velocities are imposed on the stator blading inlet and the flow through the stator is efficiently turned to the axial position and diffused through the speed of sound.

Yet another object of the present invention is to provide a high pressure ratio supersonic compressor providing a shock-in-rotor and shock-in-stator flow pattern.

Still other and further features and objects of the present invention will at once become apparent to those skilled in the art from a consideration of the attached drawings wherein two embodiments of the present invention are shown by way of illustration, wherein:

Figure 1 is a partial elevational view in cross-section through a compressor constructed according to the present invention;

Figure 2 is a fragmentary developed plan view of the rotor and stator blading of a first form of the present invention utilizing an impulse rotor; and

Figure 3 is a fragmentary developed plan view of the rotor and stator blading of a second form of the invention utilizing a shock-in-rotor and shock-in-stator blading construction.

As shown in the drawings:

In the embodiment of the invention illustrated, a rotor 10 is driven by a rotating shaft 11 mounted in a housing 12 having a generally streamlined opening annulus 13.

Blades 14 are provided radially projecting from the peripheral surface of the rotor 10 and blades 15 are rigidly secured in the housing 12 to provide a stator. The annular passage 16 leads from the trailing edge of the blades 15 to the outlet of the compressor. It will, of course, be understood that while these general components are known in the art, and as thus broadly described, are shown in the prior copending application, the specific blading which will hereinafter be described, taken in combination with the specific configuration of the inner and upper annulus walls 16a and 16b, respectively, comprise that portion for which novelty is claimed.

As described above, the major problem facing the industry in development of highly efficient compressors is the problem of efficient turning of the fluid at and above supersonic velocities. A high rate of fluid turning is extremely desirable in order to provide a high pressure ratio for a single stage and in the absence of such turning several stages are necessary in order to provide a pressure ratio on the order of 5. Accordingly, while the rotor and stator blading illustrated and described in the above mentioned copending application provide a very high efficiency as a result of its novel construction accurately controlling fluid flow contraction and positioning of the normal shock in the rotor passages, it provides a relatively lower pressure ratio per stage of rotating compressors 2.5. In the operation of the compressor set forth in the above identified copending application, it was found that an increase of fluid flow turn substantially beyond approximately 12° caused severe fluid flow separation at the exit end of the rotor blading. This flow separation and turbulence quickly dropped the overall performance of the compressor to a figure well below desirable limits and accordingly the problem of high fluid turn without such flow separation becomes the problem from which the application stems.

The present blading, having an impulse rotor, leading edge characteristics substantially identical to those set forth in the copending application, and hence is provided with a small blade included angle of approximately 4° and has provision for several increases in blade wedge angle with substantially the maximum channel contraction angle above the maximum channel contraction angle above the small blade included angle. This provides supersonic turning with several oblique shock waves upstream of a normal shock wave positioned at or slightly downstream of the blade passage throat. As a result of actual tests made on large numbers of passage shapes and contours the critical control factor in providing efficient high velocity fluid turn in supersonic blading was discovered to be the rate of area expansion provided downstream of the point of maximum channel contraction, or subsequent to the normal shock position.

Contrary to the understanding of those heretofore working in the art, effective high turn may be accomplished by limiting the area of the fluid flow channel to a very small amount. For situations wherein an inlet velocity to the rotor blading is Mach 1.6 or above it has been found that an area expansion rate of less than 5% will prevent separation of the fluid flow and hence provide efficient operation. This figure may be increased by a small area expansion rate, and where the inlet Mach number is such that a subsonic velocity, V, below Mach 1.6 without substantially affecting in an adverse way the efficiency of the blading.

This discovery, when applied to compressors provided with an inlet velocity relative to the blading, V<sub>ir</sub>, above the speed of sound or above Mach 1.6 permits high velocity fluid turn efficiently in two alternative manners. The first of these is high rate of turning without substantial decrease in velocity and accordingly substantially no change in fluid flow passage from the inlet to the outlet of the rotor blading. Such a compressor would provide a rotor operating without an impulse rotor with the velocity of the entering fluid would never drop to the speed of sound or below during its passage through the rotor. By placing the blades fairly close together to provide relatively high solidity and by providing a smoothly increasing wedge angle on the high pressure side of each blade efficient supersonic fluid flow has been accomplished without providing a plurality of strong oblique shock waves. An illustration of this arrangement is found in Figure 2 wherein the blades 14 of the rotor 19 are provided with sharp leading edges 14a followed by a curved surface forming a continually increasing wedge angle. No throat or constriction is provided in the rotor blading shown in Figure 2 and in fact, very slight area expansion may be provided in order to compensate for boundary layer build-up adjacent the surfaces of the blades. Accordingly, fluid entering the compressor along the axial line indicated by the vector V<sub>ir</sub> will have a relative velocity relative to the blading of V<sub>ir</sub> at entry into the blades and will have an exit velocity V<sub>ex</sub> relative to the blades which is substantially as great as V<sub>ir</sub> and which will be supersonic. Considering the vector diagram illustrating the velocities of the fluid entering the stator blading 15 it will be noted that the vectorial sum of the velocities V<sub>ex</sub> and V<sub>r</sub>, the latter indicating the velocity of the blades occasioned by rotation of the rotor, provides a relative velocity entering the stator V<sub>rel</sub> which is extremely high. In practice, when a rotor of the impulse type, above described, is utilized for providing a flow turn in the manner of the actual practice fluid flow turning of at least 120° is practicable.

While high fluid turning is thus provided in the im-
pulse type rotor blading above described, the extremely high velocities entering the stator increase the problem of turning the fluid flow to the axial direction and at the same time diffusing the fluid. Using the principles above discussed, it has been found that efficient stator blading may be accomplished by causing the fluid flow from its vectorial direction $V_{st}$ shown in Figure 2 to the axial direction with simultaneous diffusion through the speed of sound. This is accomplished through the provision of stator blades having a sharp leading edge and a general supersonic flow contraction pattern in both the axial and oblique shock waves. As mentioned copending application. Thus, a gradually increasing wedge angle is provided on the high pressure side of the stator blades 15, providing a plurality of oblique shock waves which diffuse the fluid flow, with some simultaneous turning, down to the speed of sound at which point the normal shock occurs. Subsequent to the normal shock the fluid flow is turned through a large angle to the axial direction subsonically with further diffusion accomplished through the provision of an expansion of the flow area downstream of the normal shock. This expansion is, as above described, limited to approximately 3% and when so limited it has been found that the fluid flow does not separate.

It will be understood that because of the oblique shocks developed during supersonic contraction of the fluid flow, through the use of a series of sharp increases in blade wedge angle, more efficient diffusion of the velocity of the fluid from supersonic to the subsonic is provided than would be provided where a single normal shock wave is developed. However, it is noted that tests and operation of compressors constructed according to the present invention show that the best total pressure recovery, or overall efficiency, of supersonic blading of the type hereunder discussed, results from operation at the highest static pressure ratio for the blades. Accordingly, positioning of the normal shock at the earliest possible point in the flow channel, with subsequent expansion and diffusion providing optimum results. This condition is somewhat in conflict with the use of a plurality of oblique shock waves since the construction of the blading to provide such waves ordinarily requires more blade length prior to the positioning of the normal shock than is necessary if the normal shock is almost immediately set up. Accordingly, it is preferred that the oblique shock waves if utilized, be set up relatively close to each other in order to move the normal shock forward as far as possible.

If space limitations should prevent the use of blading sufficiently long to provide a substantial pressure rise through diffusion, with the nominal expansion areas permitted, and it accordingly becomes necessary to choose between the efficiencies effected through the use of oblique shock wave patterns and the efficiency effected through moving the normal shock forward in the flow channel it has been found that the effect of moving the normal shock forward is superior and that viscosity and secondary flow losses are decreased more than enough for movement of the normal shock wave forward to overbalance the additional loss through elimination of the oblique shocks. Accordingly, in such an instance a rapid flow contraction adjacent to the inlet to the blades, setting up the normal shock almost immediately following the bow wave, such as for example at 20 in Figure 2 is preferred. The alternative situation, in which oblique shocks are provided is satisfactory, however, and is shown in dotted lines with the normal shock 20a lying downstream of one or more oblique shocks 20b, all lying behind the initial shock wave 21.

In Figures 3 and 4 both the rotor and stator operate transonnically, in other words, the blades of both the rotor and the stator operate to reduce the relative velocity of the fluid flowing therethrough from a supersonic relative inlet velocity to a subsonic relative outlet velocity. In order to provide a sufficiently high subsonic velocity relative to the rotor bladings at their outlet to cause a supersonic inlet velocity to the stator, it is necessary that a high angle of turning be provided in the rotor. This angle of turning need not, however, be as great in the embodiment shown in Figure 3 as in the impulse rotor arrangement shown in Figure 2 since a pressure rise occurs in the rotor as well as in the stator and accordingly a lower inlet velocity at the stator blades will provide the same overall compressor rotor and stator combination, pressure ratio. However, in view of the fact that the greater the turn provided, the greater the total pressure ratio will be, it is desired that a high degree of fluid flow turn be provided. Accordingly, the blading shown in Figure 3 is set up to provide a fluid flow turn of approximately 130° in the rotor, which is shown as the angle A between the vector quantities $V_{st}$ and $V_{ro}$. The rotor blades provide initial flow contraction, with flow turn through a plurality of oblique shock waves 14b, and subsequent expansion limited to approximately 3% as above described. The stator is constructed to diffuse and turn the fluid entering the stator blade, namely $V_{st}$ to the axial direction and accordingly, as shown, the stator blade provides a fluid turn of approximately 60° indicated by the angle B. As in the case of the rotor, the stator blades provide flow contraction through oblique shock waves 20c and normal shock wave 23, with subsequent 3% flow area expansion. The turning provided by both rotor and stator is, of course, substantially greater than heretofore considered practicable in supersonic operation and as described above has been achieved through control of the expansion of the channel area downstream of the normal shock condition. As may be seen from consideration of the vector diagrams in Figure 3, although both the vector quantities $V_{st}$ and $V_{ro}$ which together define the relative inlet velocity to the rotor and the relative outlet velocity from the rotor may be subsonic, the velocity component added by rotation of the rotor itself in the usual supersonic operating ranges of from 1200 to 1500 f.p.s. will provide a relative fluid velocity $V_{ro}$ to the rotor blading well above the speed of sound and a velocity $V_{ro}$ to the stator blading likewise well above the speed of sound. As a result of test operations it has been found that a compressor operating with both a transonic rotor and transonic stator with controlled expansion area downstream of the normal shock location, and with the normal shock location positioned in the early portion of the flow channel, as indicated at 22 and 23, respectively, an overall pressure ratio of above 4, with an overall pressure recovery or efficiency of 70% or better is accomplished.

The diagrams of Figures 2 and 3 do not, of course, illustrate structural features incorporated in the compressor for insuring radial equilibrium. It is, of course, essential for efficient operation of any compressor that the fluid flow have radial equilibrium in order to prevent excessive pressure gradients radially along the flow passages. This equilibrium may be provided as shown in Figure 1 by providing radial outward turn of the compressor flow passages. As will be noted from a consideration of Figure 1, the inner radial surface of the channel 16a moves radially outwardly at a greater rate than the outer surfaces 16b. This difference provides an apparent contraction toward the downstream portion of the blading while passing through the blades 15. This contraction in radial dimension, however, is incorporated to limit the expansion that would otherwise be caused by the necessary tapered construction of the trailing edges of the compressor blades.

The downstream of the stator, the fluid, which is traveling in the axial direction subsonically, is further diffused in an expanding channel having, as shown, an expansion rate of about 7%.

Additional radial equilibrium is, of course, obtained.
through a twisting of the individual rotor blades about their radial axes to compensate for the increase of actual blade velocity with increasing radius and to thereby provide for substantially constant effective inlet velocity \( V_{IL} \) throughout the radial length of the rotor blades. In practice this is accomplished through the provision of a decreasing angle of blade attack with increase in rotor radius. The angle of attack being the angle between the low pressure side of the individual blade adjacent its leading edge and the fluid relative inlet flow. This is demonstrated at C in Figure 2.

It will thus be apparent to those skilled in the art that we have provided a novel supersonic compressor capable of providing overall pressure ratio of approximately 4 to 1 in a single rotor-stator stage. This extremely high pressure ratio, heretofore considered impossible in supersonic compressor construction, has not only been achieved but has been accomplished while maintaining a total pressure recovery efficiency of at least 70% thereby providing an extremely practical compressor.

We claim as our invention:

1. In combination in a supersonic compressor, a rotor having a peripheral surface spaced from an annular internally facing casing surface by a plurality of radially extending rotor blades and a stator comprising an annular passage coaxial with said rotor and having a plurality of radially extending stator blades therein, first means directing inlet fluid to said rotor blading at a velocity relative thereto above Mach 1, said rotor blades having a turn in excess of approximately 10° along their axial length and a degree of fluid channel area expansion in the downstream direction less than 3%, whereby fluid entering said rotor leaves said rotor traveling at a velocity relative to said stator above a velocity of Mach 1 and having a vectorial movement in the same direction as the rotation of said rotor, said stator having a blade inlet relative velocity above Mach 1 and said stator blades likewise having a turn in excess of approximately 10° along their axial length whereby the fluid leaving said rotor blades is turned to the axial direction by flow between adjacent blades, second means associated with said stator for substantially simultaneously turning and diffusing said flow subsonically without introducing flow separation, said last named means including surfaces on adjacent stator blades and said stator inner and outer annular walls providing a cross-sectional area expansion in the downstream direction from a point of minimum area to the outlet of said stator blade passageways on the order of 3%.

2. In combination in a supersonic compressor, a rotor having a peripheral surface spaced from an annular internally facing casing surface by a plurality of radially extending rotor blades and a stator comprising an annular passage coaxial with said rotor and having a plurality of radially extending stator blades therein, first means directing inlet fluid to said rotor blading at a velocity relative thereto above Mach 1, said rotor blades having a turn in excess of approximately 10° along their axial length and a degree of fluid channel area expansion in the downstream direction less than 3%, whereby fluid entering said rotor leaves said rotor traveling at a velocity relative to said stator above a velocity of Mach 1 and having a vectorial movement in the same direction as the rotation of said rotor, said stator having a blade inlet relative velocity above Mach 1 and said stator blades likewise having a turn in excess of approximately 10° along their axial length whereby the fluid leaving said rotor blades is turned to the axial direction by flow between adjacent blades, second means associated with said stator for substantially simultaneously turning and diffusing said flow subsonically without introducing flow separation, said last named means including surfaces on adjacent stator blades and said stator inner and outer annular walls providing a cross-sectional area expansion in the downstream direction from a point of minimum area to the outlet of said stator blade passageways on the order of 3%.
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fluid to said channels at a velocity above Mach 1 relative thereto, means on adjacent blades for contracting said fluid flow and thereby diffusing said flow to a velocity of Mach 1 and means associated with said blades and said walls downstream of the point of maximum fluid flow contraction for simultaneously causing fluid flow turn in excess of approximately $10^5$ and an expansion thereof on the order of 3%.

7. In a supersonic compressor, a pair of blades forming in combination with a pair of annular wall surfaces, a flow channel, means introducing fluid to said channel at a relative inlet velocity above Mach 1, means on the opposing faces of said blades for contracting said fluid flow at the maximum rate said flow may be contracted without blocking said flow to thereby position the normal shock resulting from said contraction at a point adjacent the leading edges of said blades, and means on said opposed blade faces and said walls downstream of the point of maximum fluid flow contraction for simultaneously turning said fluid flow through an angle in excess of approximately $10^5$ and expanding said flow at a rate on the order of 3%.

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