Aug. 16, 1960 V. H. PAVLECKA 2,949,224

SUPERSONIC CENTRIPETAL COMPRESSOR

Filed Aug. 19, 1955 7 Sheets-Sheet 1

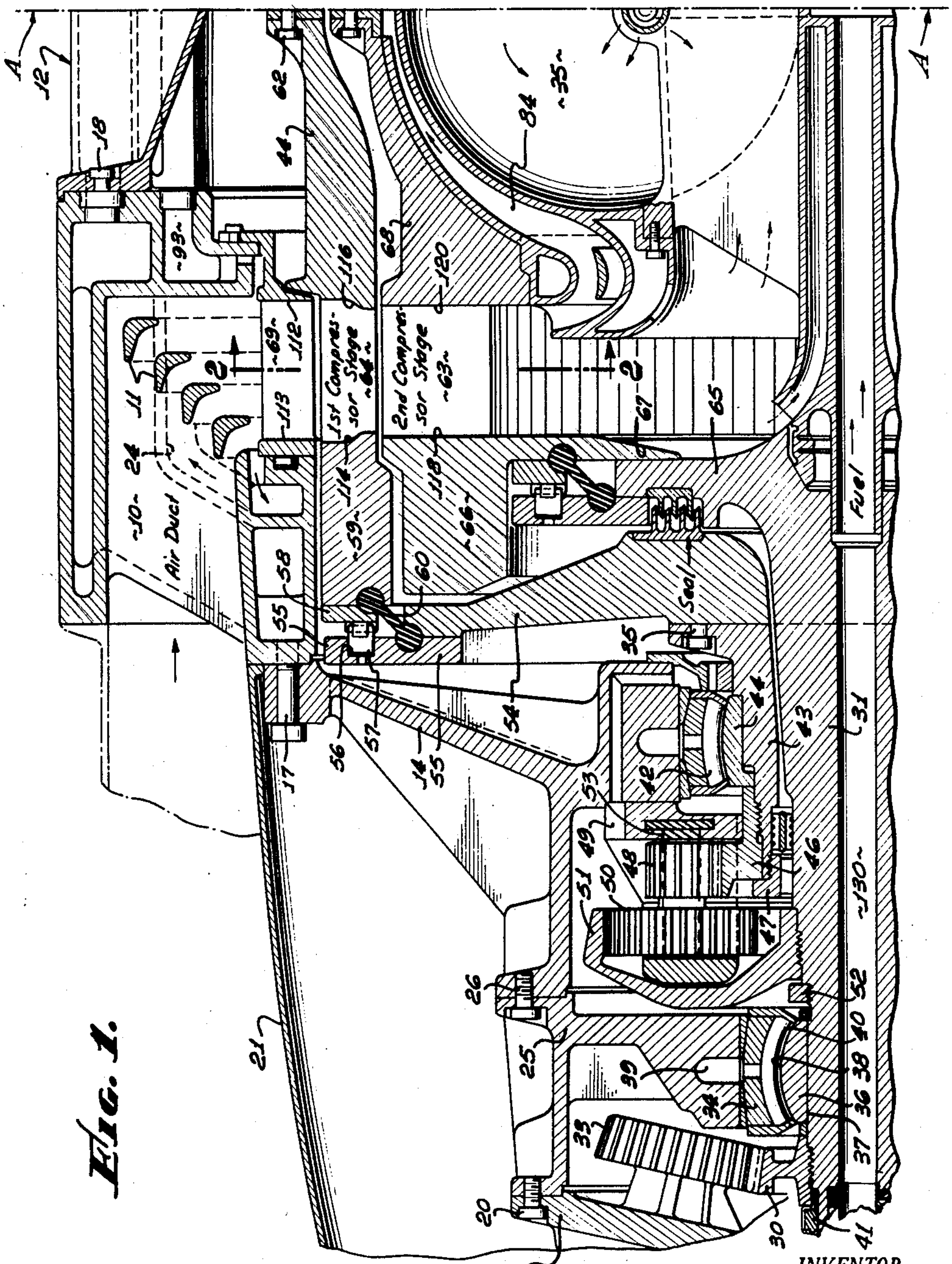

FIG. 1.

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SUPERSONIC CENTRIPETAL COMPRESSOR

Filed Aug. 19, 1955 7 Sheets-Sheet 2

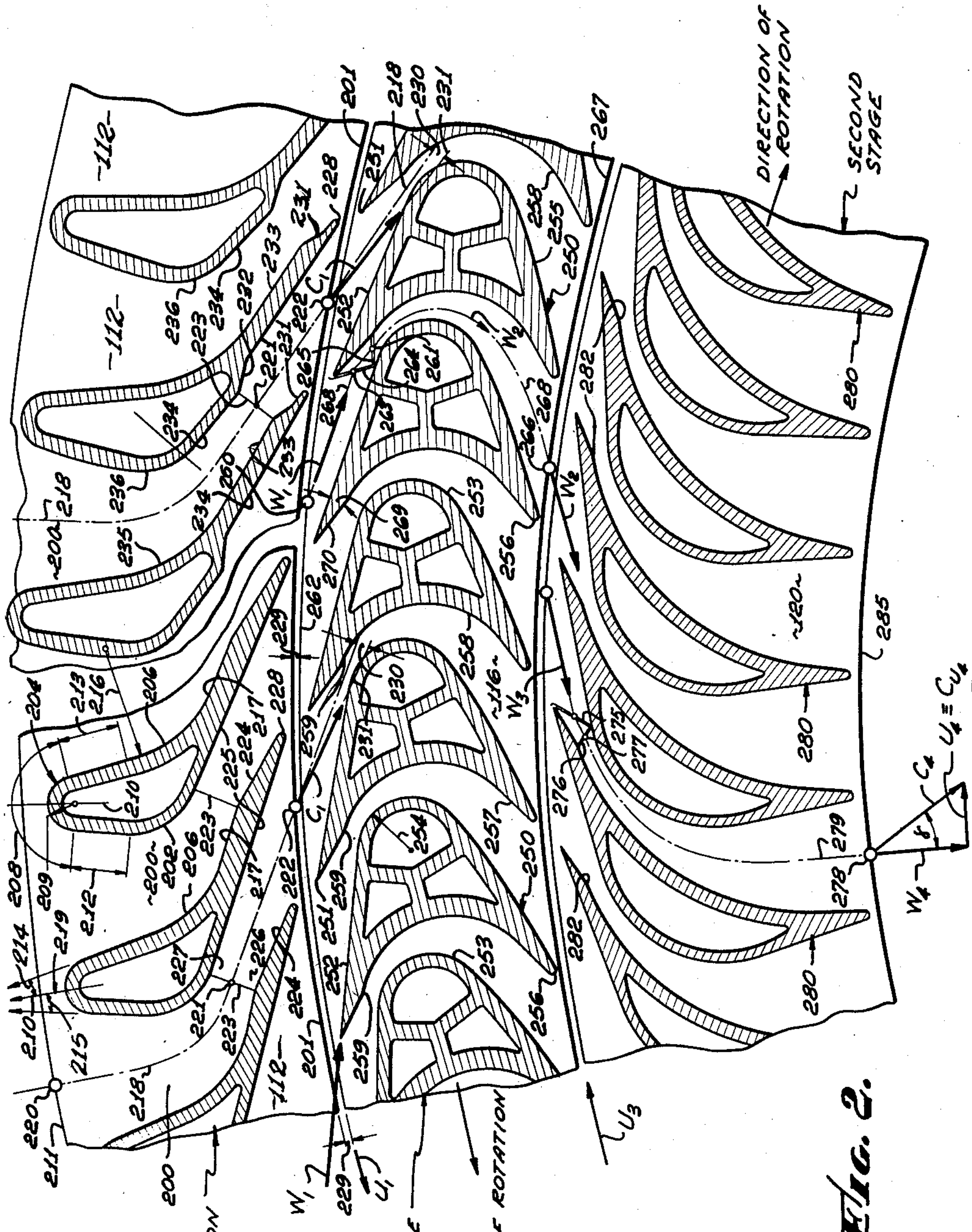

FIG. 2.

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SUPERSONIC CENTRIPETAL COMPRESSOR

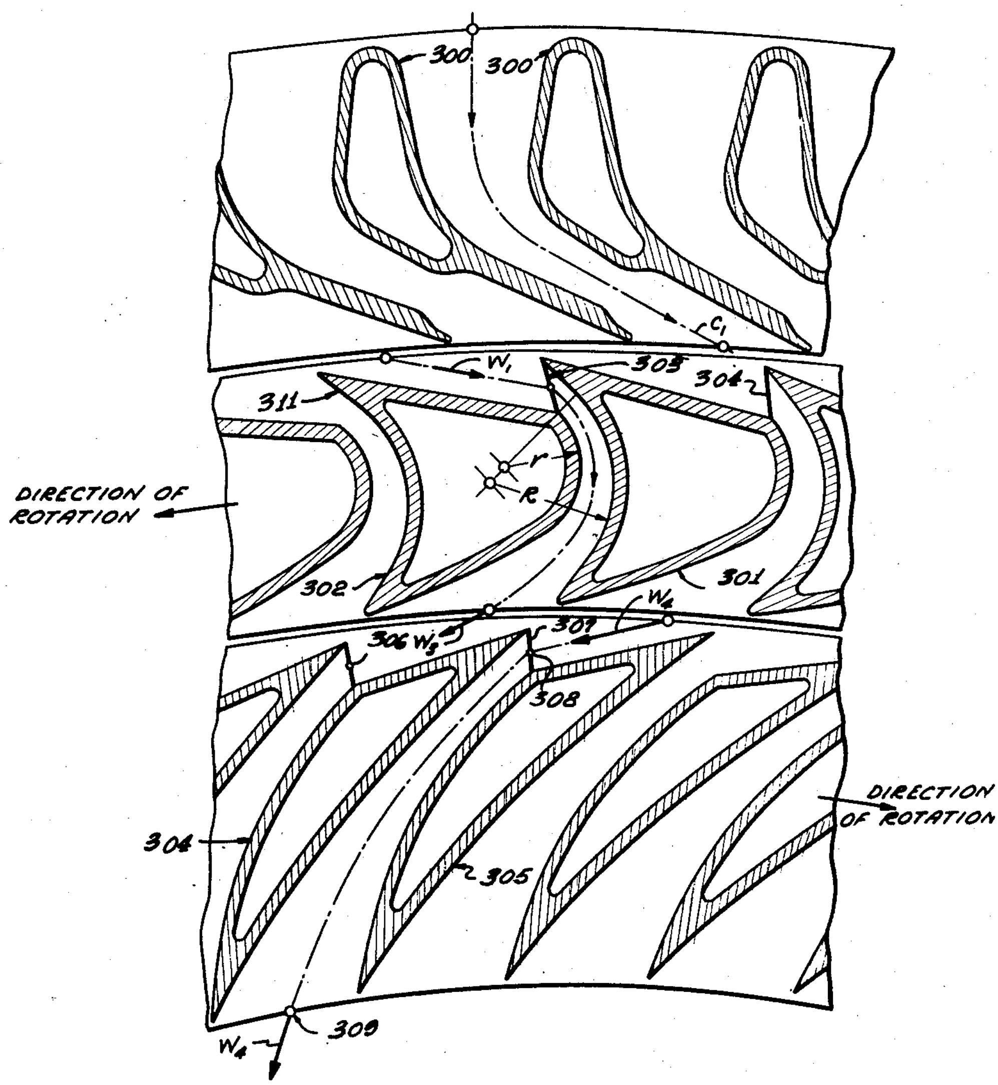

FIG. 3.

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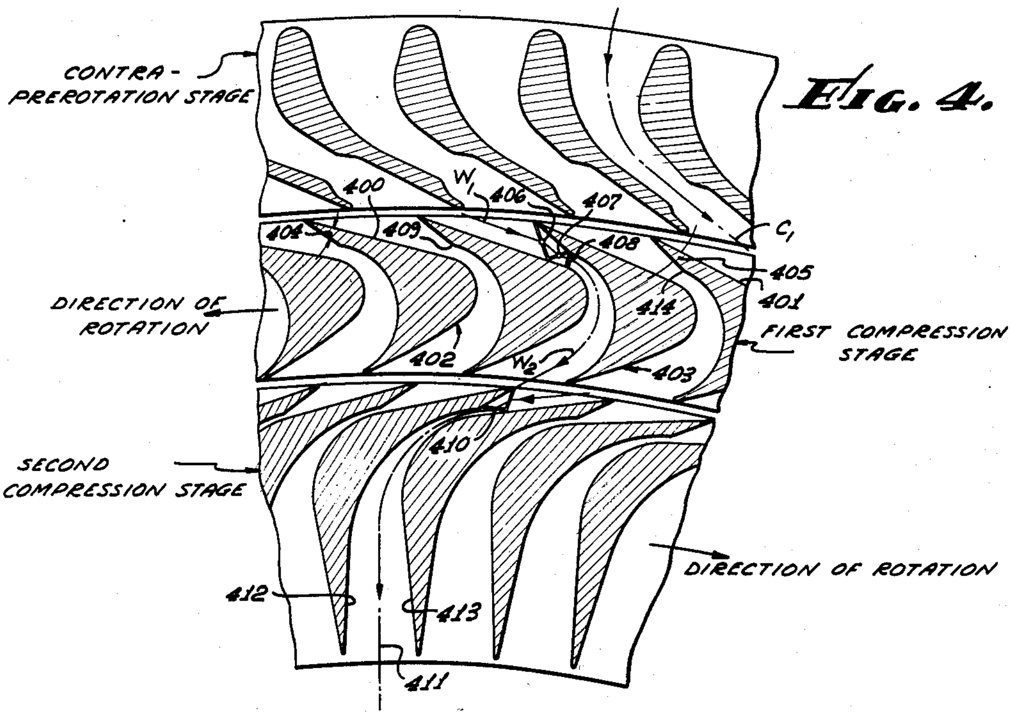

FIG. 4.

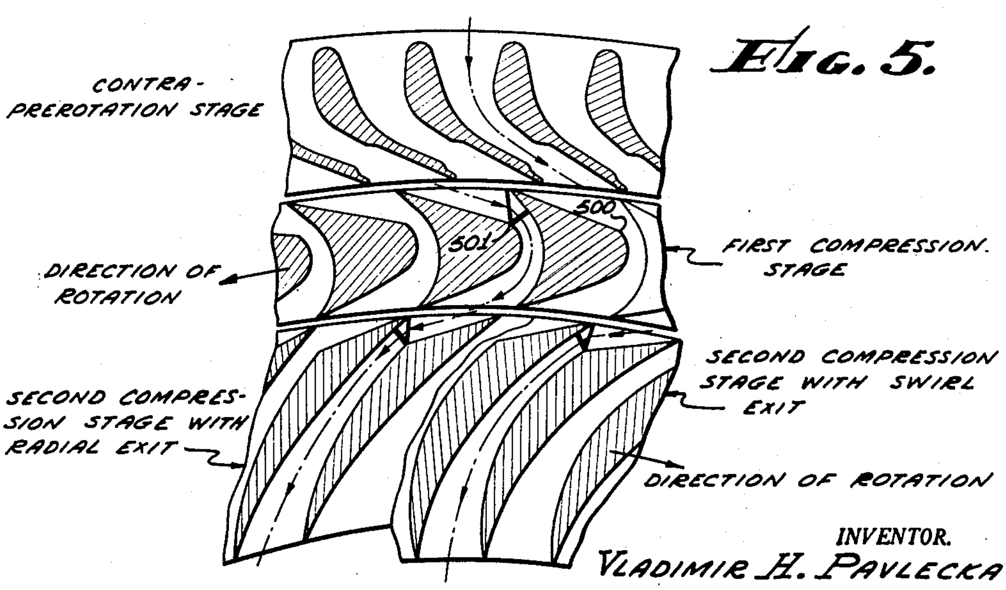

FIG. 5.

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SUPERSONIC CENTRIPETAL COMPRESSOR

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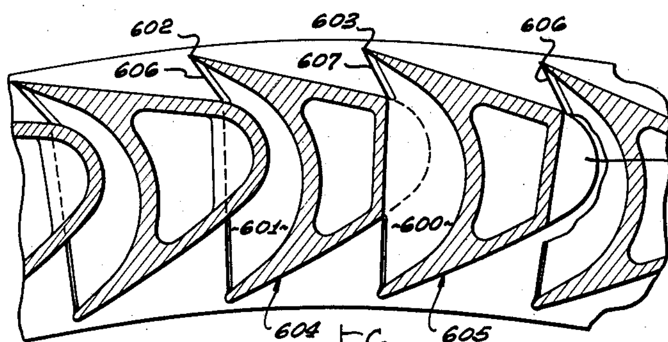

FIG. 6.

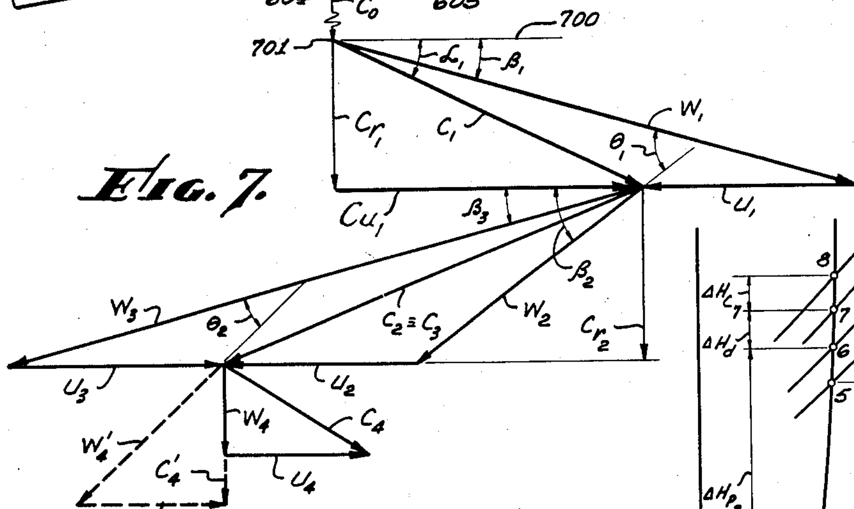

FIG. 7.

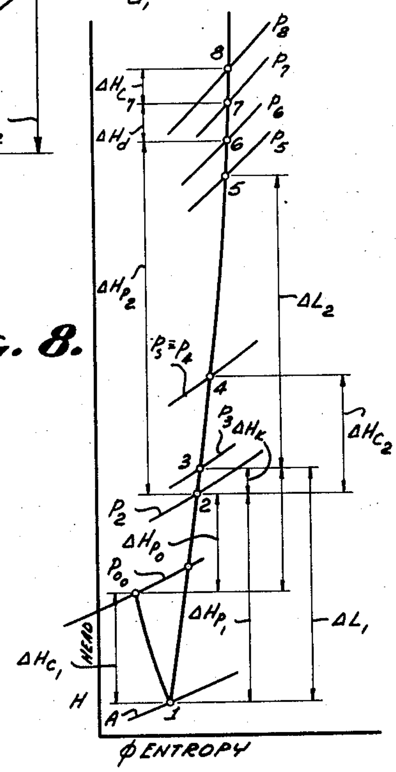

FIG. 8.

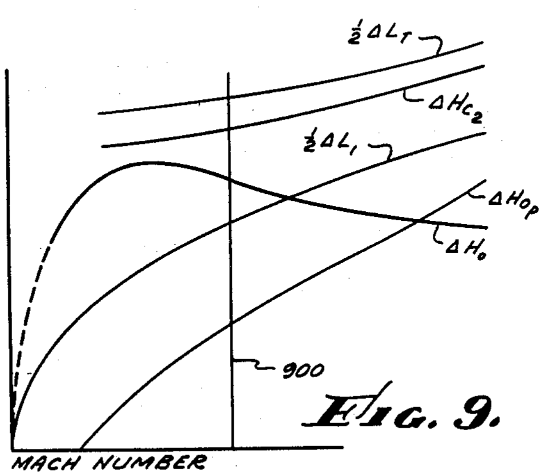

FIG. 9.

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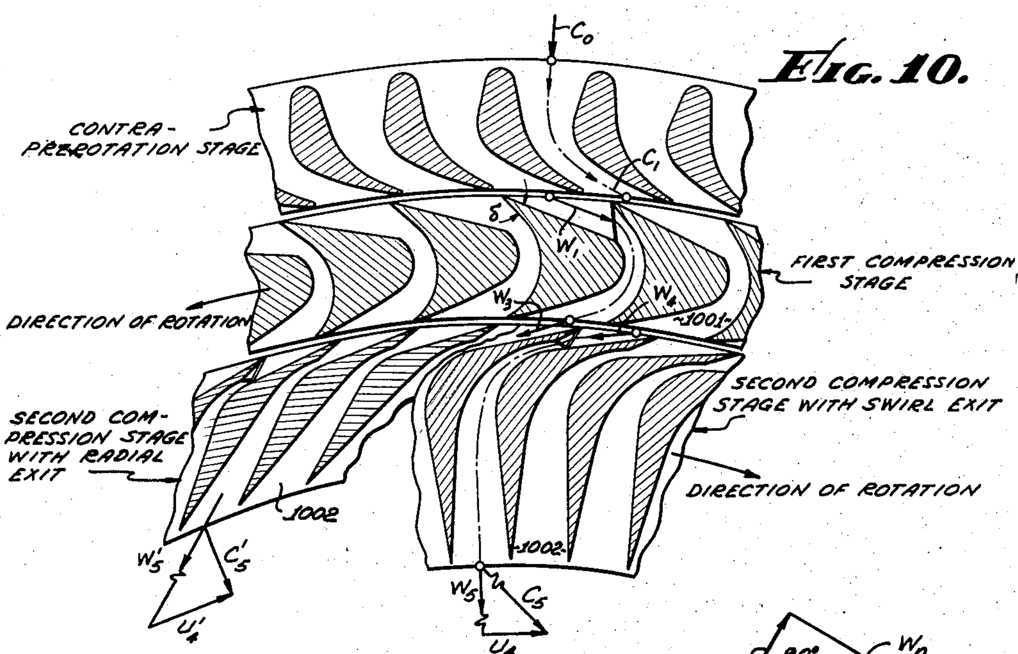

FIG. 10.

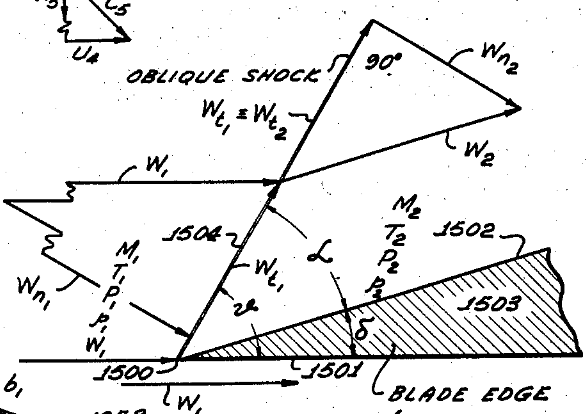

FIG. 15.

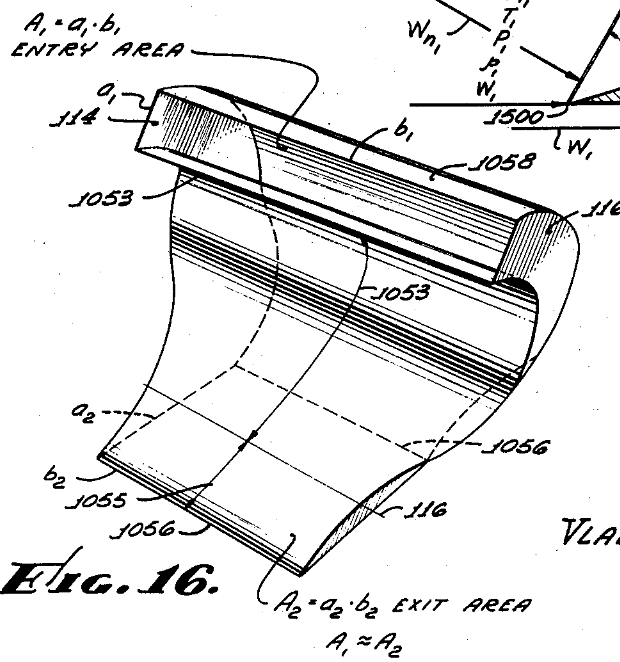

FIG. 16.

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SUPERSONIC CENTRIPETAL COMPRESSOR

Filed Aug. 19, 1955 7 Sheets-Sheet 7

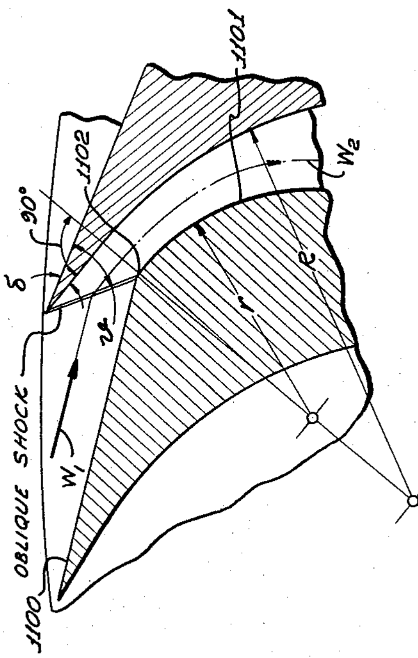

Fig. 11.

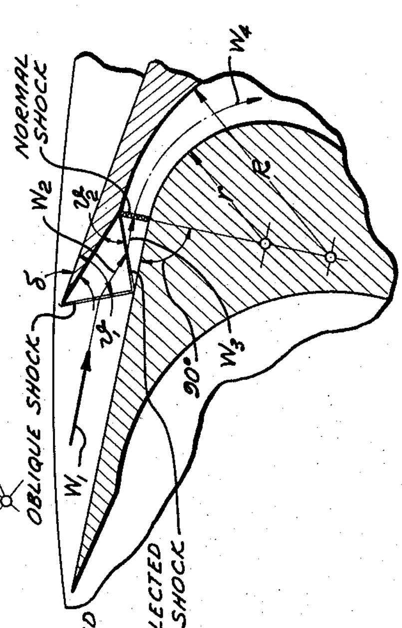

Fig. 14.

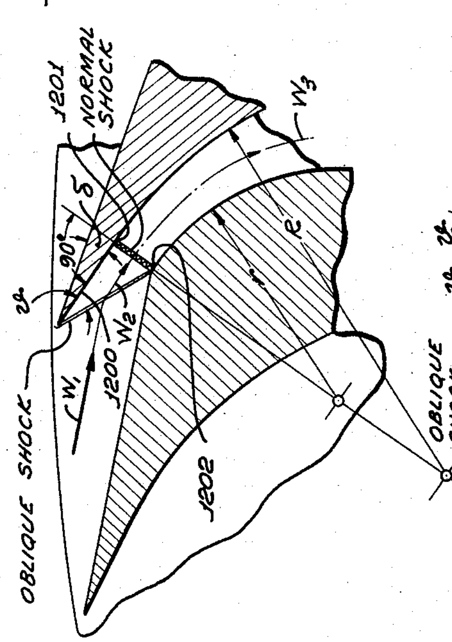

Fig. 12.

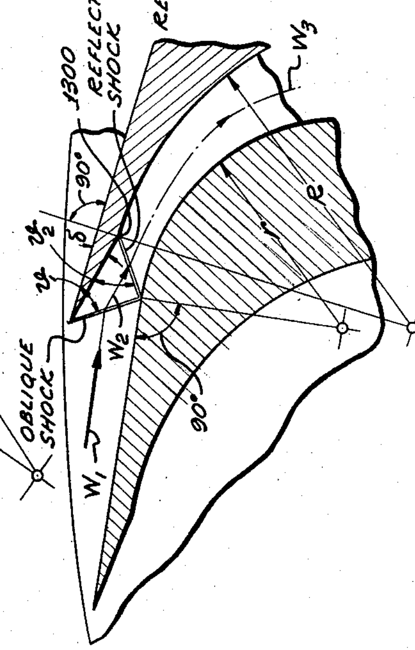

Fig. 13.

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United States Patent Office

2,949,224

Patented Aug. 16, 1960

2,949,224

SUPERSONIC CENTRIPETAL COMPRESSOR

Vladimir H. Pavlecka, Pacific Palisades, Calif., assignor to American Machine & Foundry Company, a corporation of New Jersey

Filed Aug. 19, 1955, Ser. No. 529,504

25 Claims. (Cl. 230—124)

This invention relates to dynamic compressors of centripetal flow type, and more particularly to the supersonic centripetal flow compressors.

This application for patent is a continuation-in-part of the parent application for patent, #217,347, filed March 24, 1951, entitled Gas Turbine Power Plant, now U.S. Patent No. 2,804,747, issued September 3, 1957, which discloses and claims the entire gas turbine power plant. The supersonic compressor is disclosed and claimed in this application. The supersonic compressors having more than two stages and including a vector adjusting stage, originally disclosed in the parent case, is also disclosed in the continuation-in-part case Serial Number 514,001, entitled Methods of Compressing Fluids With Centripetal Compressors.

It is an object of this invention to provide a novel centripetal supersonic dynamic compressor in which compression is accomplished by means of oblique, reflected, and normal shocks singly or in any combination with each other, as well as subsequent subsonic diffusion, the oblique shock being produced by a leading flat surface of a blade.

Still another object of this invention is to provide a novel centripetal compressor having a prerotation stage and a plurality of contra-rotatable rotor stages, all rotatable compressor stages being supersonic on the input side and subsonic on the output side.

Still another object of this invention is to provide means for synchronizing the rotation of the contra-rotatable compressor rotors of a supersonic centripetal compressor for maintaining proper velocity vector relationships for compressed air flowing between the compressor stages.

It is also an object of this invention to provide new methods of compressing fluids by accelerating the fluid either to a supersonic or subsonic velocity and then compressing it by producing a single or a plurality of supersonic compression shocks by means of wedge-shaped blades, the lagging surface of the blades being parallel to the relative velocity of the fluid at its entry into the compression stages, the methods being applicable to the centripetal, centrifugal and axial methods of compressing gases.

Yet another object of this invention is to provide a supersonic centripetal compressor in which a stationary contra-prerotation stage is provided for accelerating a fluid to be compressed either to a subsonic or supersonic velocity at the exit from the contra-prerotation stage and then subjecting the accelerated fluid to a plurality of supersonic acoustic shocks in the succeeding contra-rotatable supersonic compression stages.

The novel features which are believed to be characteristic of the invention, both as to its organization and method of operation, together with further objects and advantages thereof, will be better understood from the following description taken in connection with the accompanying drawings in which several embodiments of the invention are illustrated as examples of the invention.

It is to be expressly understood however, that the drawings are for the purpose of illustration and description only, and are not intended as a definition of the elements of the invention. Referring to the drawings:

Figure 1 is an axial vertical sectional view of a two-stage centripetal supersonic compressor.

Figure 2 is a transverse cross-sectional view taken along line 2—2, Fig. 1, of the contra-prerotation stage and two compression stages of the centripetal supersonic compressor.

Figures 3, 4, and 5 are similar cross-sectional views of the modified versions of the centripetal supersonic compressors disclosed in Figs. 1 and 2.

Figure 6 is an enlarged transverse cross-sectional view of compressor blades and blade reinforcing ribs.

Figure 7 is a vector diagram of fluid velocities in a two-stage centripetal supersonic compressor.

Figure 8 is the entropy-head diagram of the compression process obtained in the disclosed compressors.

Figure 9 illustrates a family of performance curves of the first supersonic compression stage as a function of the local Mach number.

Figure 10 is a transverse cross-sectional view of a centripetal compressor with a subsonic contra-prerotation stage and two supersonic compression stages.

Figures 11, 12, 13, and 14 are transverse sections, taken in a plane perpendicular to the axis of rotation of the compression stages, of the input portion of the compression stages and particularly, leading portions of the blades and of the flow channels formed by the opposing surfaces of the adjacent compressor blades.

Figure 15 is a side cross-sectional view of a leading edge of a compressor blade and supersonic air flow over such portion of the blade.

Figure 16 is a perspective view of a flow channel of the first, rotatable compression stage.

Referring to Fig. 1, it discloses the axial vertical sectional view of the compressor portion of the gas turbine power plant and of a portion of the rotatable toroidal combustion heat generator which interconnects the second compressor stage with the first turbine stage not illustrated in Fig. 1. For a more detailed description and illustration of the entire power plant and especially of the turbine half of the power plant, reference is made to the previous mentioned parent application #217,347, now Patent 2,804,747, illustrating and describing the entire gas turbine power plant. The power plant is mounted on a composite frame consisting of an air intake duct portion **10**, a mid-portion member **12**, a front member **14**, and a rear member comparable to the front member **14**, which houses and supports the compressor and turbine portions of the power plant not illustrated in Fig. 1. The several frame members are peripherally bolted together at **17**, **18**, **26**, and **20** so as to constitute a unitary frame. The front end of the plant is cowled in a front cowl **21**. The front member **14** is extended forward by having a ring-shaped member **25** bolted to it by means of a peripheral joint held together by a plurality of bolts **26**. The ring-shaped frame member **25** is closed off by means of a slanted flat ring **27**. The compressor-turbine structure includes an inner shaft **31** mounted on two radial-axial bearings such as a bearing **34**, only one bearing being visible in Fig. 1. The second bearing is mounted on the extension of shaft **31** on the turbine side of the power plant which does not appear in Fig. 1. A similar bearing **42** is used for mounting shaft **43** which supports the outer side disc **54** and the first stage **64** of the compressor and the second stage of the turbine. The first stage **64** of the compressor constitutes a part of the outer cylinder including ring **59** and a central cylindrical member **44** connected to the identical cylindrical member located on the other side of line A—A, which bisects the power plant into two halves through the center of the toroidal combustion heat generator **35**. The cylindrical member **44** is fastened to the identical cylindrical member on the turbine side of the power plant by means of a plurality of bolts **62**. Ring **55** is provided with a plurality of cylindrical recesses **56**, which house a corresponding plurality of cylindrical pins **57**, forming a sliding fit with the cylindrical recesses. The opposite ends of the pins **57** form radially sliding fit with the corresponding radial slots in the ring **58**, the entire assembly acting as a torque-transmitting means and also as an expansion joint. Ring **58** is bolted to a substantially rectangular ring **59** which constitutes the outer hoop ring of the first compression stage **64** of the compressor. The inner side-surface of the hoop ring **59** is used for mounting the blades of the compressor all the way around the periphery of this ring. Rings **58** and **59** are made of titanium alloy which permits higher peripheral velocities than those obtained with steel rings. Side discs **54** and ring **55** on one side, and ring **58** and hoop ring **59** on the other side, are interconnected by a pivoted ring **60** having a cross section identical to that of a dumbbell, the two cylindrical surfaces of ring **60** forming a sliding seat within the respective seats. When the rings **58** and **59** expand radially, their expansion exceeds radial expansion of side discs **54** and this differential expansion is transmitted and absorbed primarily by the elastic deformation of ring **60**. The right sides, as viewed in Fig. 1, of the blades **64** of the first compression stage are welded to the left side-surface of an outer torque-transmitting hollow cylinder **44** composed of two rings forming a central bolted joint **62**. The right side-surface of the hollow cylinder, including member **44**, terminates in the plurality of blades in the second stage of the radial centrifugal flow tubine, which is not illustrated in the figure.

The left portion of shaft **31** is provided with an integral side disc **65**, which forms a similar expansion joint with a hoop ring **66** provided with a fluid-accelerating surface **67** located in the exit channel of the compressor.

Shaft **31** is mounted in two bearings, such as a bearing **34**, made preferably of bearing aluminum alloy. The front end bearing **34** consists of a steel ring **36**, the flat inner surface **37** of which forms a sliding axial fit with shaft **31** but is keyed to the shaft to prevent its rotation around the shaft. The outer surface **38** of ring **36** has a spherical surface constituting the sliding surface for the similarly shaped sliding surface of bearing **34**. The frame member **25** is provided with an oil gallery **39** connected to a source of lubricating oil, this gallery communicating with oil grooves **40** which run parallel to shaft **31**. Means are also provided for sealing the entire periphery of bearing **34** on its sides to prevent excessive sideway oil leakage from the grooves **40**. The position of the bearing ring **36** is fixed on shaft **31** by an end nut **41** so that the engagement of the two spherical surfaces fixes the longitudinal position of shaft **31** with respect to frame member **25**. Similar bearings **42** are used for mounting shaft **43** by means of a spherical ring **44**, ring **44** being provided with a shoulder for longitudinal location of shaft **43**. This longitudinal location is obtained through a ring gear **46** which is held in fixed position by a ring nut **47** and circumferential splines between shaft **43** and the ring gear **46**. Ring gear **46** is geared to a plurality of circumferentially positioned pinions **48** which revolve around their respective shafts such as shaft **53** mounted in a ring resembling somewhat a squirrel cage solidly bolted to a frame member **14**. The squirrel cage ring **49** is provided with four large openings which house four synchronizing gear assemblies including gear ring **46**, pinion **48**, gear **50**, and a ring gear **51** provided with gear teeth on its inner surface. The ring gear **51** is splined to shaft **31** and is held in a fixed axial position by a ring nut **52** threaded to a shaft **31**. The synchronization of the two shafts is obtained by transmitting any differential torque between the shafts **31** and **43** through the synchronizing gears **46**, **48**, **50**, and **51**. Pinion **48** and gear **50** constitute one solid piece mounted on the same shaft **53**. Since shafts **53** prevent the rotation of the synchronizing pinions **48** and **50** around the two gear rings **46** and **51**, the gear assembly will act as a synchronizing means between the shafts **31** and **43**.

The second stage of the compressor is welded on one side to the hoop ring **66**, and on the other side to a torque transmitting hollow cylinder **68** similar to the hollow cylinder **44**.

Ambient air enters the air duct **10**, passes through the compressor contra-prerotation stage **69**, the rotatable compression stages **64** and **63**, and then enters the toroidal combustion heat generator **35**, which is described more fully in the parent case 217,347, now Patent 2,804,747, and divisional application S.N. 606,451, filed August 27, 1956. The combustion chamber revolves with the second stage **63** of the compressor and the first stage of the turbine.

The compressed air which by-passes the combustion heat generator and the heated gases from the combustion heat generator enters a radial input duct into a radial centrifugal flow turbine having first and second radial stages mounted in the same manner as the compressor stages. These two contra-rotating turbine stages rotate in opposite directions the two stages of the compressor, the torque developed by the turbine stages being transmitted to the compressor stages through the hollow cylinders **68** and **44**. The two contra-rotating turbine stages are rotating at two different angular velocities, the ratio between these two angular velocities being held constant by the synchronizing gear train between the two rotors of the turbine and the two rotors of the compressor. As will be explained more fully later in this specification, proper vectorial relationships are maintained in the compressor and the turbine by maintaining the angular velocities ratio constant between all rotating elements of the power plant. The angular velocity of the compressor is controlled by a governor geared to a ring gear **30** which controls the rate of fuel supply furnished to the combustion chamber. This, in turn, controls the temperature of gases leaving the heat generator, thereby controlling the angular speed of the turbine and compressor. The governor, therefore, does not maintain this speed constant but adjusts it to maintain a constant Mach number throughout the plant, which is especially desirable because of the supersonic type of compressor.

The joint application of the present applicant and Frederick Dallenback, filed August 12, 1950, Serial No. 179,028, now Patent No. 2,712,895, entitled "Centripetal Subsonic Compressor," discloses a centripetal subsonic compressor. The following appears in the above application under subtitle "Fluid Dynamics and Thermodynamics of the Centripetal Compressor": "In any rotary compressors having continuous compression cycle, as differentiated from intermittent cycle (reciprocating compressors), the only way that the process of compression can be accomplished is by first accelerating an elastic fluid to as high velocity as possible to obtain as high a kinetic energy as possible from the ambient potential energy state and then converting or transforming this kinetic energy into potential energy (pressure)." The above statement leads one to an immediate conclusion that the higher is the velocity of the accelerated air prior to diffusing or shock conversion of the kinetic energy into pressure (if air is the fluid medium), the higher will be the final pressure obtained with the compressor. This being the case, the next logical question is whether there is any practical limit to this velocity. Such absolute limit does exist, and it is within the supersonic velocity range. The absolute velocity limit is determined by the overall gain in compression obtained in the first rotor stage which is also a function of the Mach number of the absolute velocity at the entry to the first compression stage. When this Mach number reaches a value of the order of 1 to 1.3, the gain in the first compression stage is maximum. Beyond 1.3, the absolute pressure in the gap between the prerotation stage and the first compression stage becomes so low that the first compression stage cannot overcome this drop in pressure with any additional increase in compression. Stated differently, the higher the Mach number in the gap, the lower is the gap pressure. Therefore, the first stage before producing any gain must overcome lower and lower pressure as the Mach number increases. There comes a point at which any further increase in the Mach number begins to reduce the overall compression ratio. This will be discussed more fully in connection with the compression ratio curve illustrated in Figs. 8 and 9. It is one of the requirements of the supersonic compressors that, in order to operate supersonic compressors at optimum performance characteristics, it is necessary to operate such compressors at substantially constant Mach number. In some applications, it is desirable to operate the compressors at variable speed, and when this is the case, then it becomes more advantageous to use subsonic compressors whose characteristics permit their operation at variable speeds without materially affecting their performance. The disclosed compressor is a supersonic compressor.

Irrespective of whether the compressor is subsonic or supersonic, the maximum obtainable compression ratio is a function of the maximum peripheral velocity which can be attained safely with the existing high strength alloys. As illustrated in Fig. 9, the higher is the peripheral velocity, the higher is the maximum compression ratio obtainable in the first rotor. Therefore, one of the factors limiting this ratio is the maximum allowable proportional limit of metals used for making the compressor rotor. For conventional steel, this limit is reached when the peripheral velocity is of the order of 700 feet per second in centripetal compressors with ring rotors, while with the newly developed titanium alloys this limit can be extended up to 850 feet per second. It is even higher for the new vacuum-processed steels. Supersonic compression of gases is accomplished by means of aerodynamic shocks induced in the channels formed by the blade surfaces. The sharp entry edge, and particularly the leading or pressure side of the rotor blade is used for producing an oblique shock and the converging compression channel defined by the opposing surfaces of two adjacent blades produces a normal compression shock. There also may be a reflected shock following the oblique shock. Adiabatic compression of gas raises its temperature and its sonic velocity; therefore, the supersonic compressor, after producing a series of shocks and the concomitant supersonic shock compression, becomes a subsonic compressor where additional compression is obtained through additional subsonic diffusion of the compressed fluid.

Referring to Fig. 2, it illustrates a cross-sectional view, normal to the axis of rotation, of the prerotation stage and two supersonic compression stages.

Prerotation stage of any dynamic compressor must expand the ambient air pressure to a sub-atmospheric pressure thereby accelerating the sttationary ambient air mass to a velocity whose vector is acceptable to the first stage of the compressor. The relative entry velocity, into the first stage, which is known in the art as velocity W_1, should be as high as possible. In the disclosed supersonic centripetal compressor, the Mach numbers

$$\frac{W_1}{a_1}=\text{Mach number} \qquad (1)$$

where a_1 is the local velocity of sound) of the relative velocity W_1 can reach values of the order of 1.5 to 3.0, depending on the fluid used. Therefore, the configuration of the entry channel, which is the channel defined by the convex surface **202** of the prerotation blade **204** and by the concave surface **206** of the identical blade **207** must have a pre-acceleration region for gradual acceleration of the air mass crossing the plane defined by the outer periphery of the prerotation stage, and then the acceleration region which enables one to each the relative velocity in the input of the first compression stage having a Mach number of the order of 1.5 to 3.0. The pre-acceleration region is necessary to avoid first, in the first half of the entry region, excessive friction losses because of large area exposed to the flow, and second, to avoid flow separation in the intermediate region of the channel where maximum turning of the flow occurs and where it blends into the accelerating channel proper.

The blades, therefore, assuming the shape of curved or cambered blades which begin with a circular leading edge **208** having a radius **209** whose center is positioned on the radial line **210** of the entire compressor circle **211**. The magnitude of radius **209** is not critical and is primarily dictated first by the ability of the cylindrical surface to divide the air flow equally into each pre-acceleration flow channel, and secondly, to avoid flow separation in the process, the latter consideration setting the maximum length that can be assigned to radius **209**.

The cylindrical leading edge **208** of the blade then blends into two sloping planes **212** and **213** on the opposite sides of the cylindrical leading edge, these planes forming two equal angles, **214** and **215** with the radial line **210**. The magnitude of these angles is of the order of 8° to 12°. The right side of the blade **204**, as viewed in Fig. 2, then blends into a circle having a radius **216** whose magnitude is not especially critical and is primarily determined by the rate of acceleration desired in the subsonic acceleration region of the flow channel. The shape of the flow channels is defined by the flat inner side-surfaces **113** and **112** (see Fig. 1) and by opposed surfaces of the blades. The flow channel will have a curved median axis line **218** bisecting arc **219** (an arc of outer periphery **211** between radial lines **210**) at point **220** at the entry into the channel. Axis line **218** also bisects a line **223**, the length of which is equal to the minimum width of the flow channel between the surfaces of the blades. Line **223** constitutes the boundary line between the subsonic and supersonic acceleration regions of the entire channel. Beyond line **223** the channel assumes the shape of a supersonic expansion nozzle defined by two straight surfaces **224** and **225** which form equal obtuse angles **226** and **227** with line **223**. The median line **218**, between points **221** and **222**, is a straight line and is normal to line **223**. The supersonic expansion nozzle terminates at the inner periphery **201** of the prerotation stage where all blades terminate in trailing edges **228**. From a purely theoretical point of view these edges should be razor-sharp edges to avoid any excessive wakes if one is to consider the entire problem of fluid dynamics only in terms of ideal flow. However, mechanical strength and stiffness requirements make it necessary to end the trailing edges as cylindrical surfaces having a small radius. This, obviously, produces a small degree of turbulence, the overall effect of which is beneficial rather than detrimental if not carried to an extreme. These small wake turbulences diminish the rate of growth of the thickness of the boundary layers in the succeeding compression stage, which would be far more detrimental than the small energy loss due to the wakes. To utilize this beneficial wake effect in the succeeding stage, the air gap **229** should be made as small as mechanically possible.

The inner end of the arc produced by radius **216** forms a tangent connection with a substantially straight line **217** which extends to line **223** where it makes a junction with line **224** of the supersonic nozzle. Line **225** terminates at line **223** at which point it is tangent to arc **202**. Arcs **202** and **206** may be circular arcs or may be complex curves; the only requirement is that no separation is introduced by these surfaces, and the general proportioning of the channel is such that the maximum rate of acceleration in the subsonic region of the channel is delayed sufficiently to avoid needless frictional losses in the early part of the channel. The blades can easily be made hollow to decrease the weight of the compressor.

The mean flow line **218** stops at point **222**, at the inner periphery of the contra-prerotation stage, and then it is deviated by an angle **230**, which is known as the flow deviation angle. This angle is formed by two lines **218** and **231**, line **231** designating the direction of actual flow after the exit from the prerotation stage, and therefore constitutes the direction of an actual velocity vector C_1 of the fluid flow in the interstage gap **229**. Therefore, in order to obtain proper vectorial relationship with respect to the first rotor, the mean flow line **218** of the prerotation stage must be turned by the same angle in a counterclockwise direction direction as viewed in Fig. 2. Stated differently, angle **230** thus becomes the correction angle for the geometry of the supersonic region of the prerotation stage.

Fig. 2, on the right side, also illustrates another version of the prerotation stage in which the supersonic nozzle of the straight type is replaced with a supersonic nozzle in which the acceleration is produced in a shorter distance by means of two curved surfaces **231** and **232** originating at the minimum width of the flow channel. The advantages of this nozzle are: (1) shorter length and therefore, smaller loss due to friction; (2) more uniform velocity distribution across the adjacent portion of the subsonic region; and (3) greater ease of introducing the correction angle **230**. The first advantage results from the shorter length of the nozzle because the expansion is more rapid. The second is due to the fact that lines **233** and **234** are substantially straight lines which form equal angles with the median line **218**, which is also a straight line at this part of the channel. Therefore, equal degrees of convergence of the two channel surfaces will produce equal degrees of acceleration. Hence, except for the boundary layer, there is a uniform velocity distribution. These surfaces, together with the viscosity effect, will tend to equalize the flow velocities even though these velocities are not uniform at the plane of entry of fluid into this region because of the velocity differences produced by the surfaces **235** and **236**. A further refinement may be introduced into this region by making the angle of line **233** with respect to line **223** smaller than the angle of line **234** to make the velocity distribution, except for the boundary layer, exactly uniform at line **223**.

The first supersonic compression stage consists of hollow blades **250** having sharp leading edges **251**. To avoid excessive wear, vibration and bending, the leading edges are actually cylindrical surfaces having small radii. Surface **252** of the blade is a flat surface parallel to the relative velocity vector W_1. It will be referred, at times, as the lagging surface of the wedge-shaped portion of the blade. It terminates as a tangent to a preferably cylindrical surface **253** having a radius **254**. Surface **253** merges into an arcuate surface **258**, which terminates at an edge line **259**. The remaining surface **270** of the blade, from edge **259** to the cylindrical tip **251**, is a flat surface, and it will be referred to, at times, as the leading surface of the wedge portion **270—251—252** of the blade. The channel consists of a compression channel from a point **260** to a point **261**. Point **260** is on the outer periphery **262** of the compression stage, and the position of point **261** is determined by the position of the boundary between the supersonic compression portion of the flow channel and the subsonic, substantially constant velocity channel, this boundary being defined by the position of the reflected shock plane **265**. Point **263** is a median point on the line defining the position of an oblique shock plane **264**. From point **261** to point **266**, the latter lying on the inner periphery **267** of the compression stage, the channel is a subsonic duct whose normal-to-flow cross-sectional areas are made constant to maintain constant flow velocity. The median flow in the channel is designated by the median flow line **268** which joins the above mentioned points. The sharp edges **251** and the leading surface **270**, or the pressure side of the blades encounter the fluid moving against them with the relative velocity W_1. This produces an oblique shock, the latter producing a stationary wave front **264** or an oblique compression wave existing in the plane defined by an "infinite" number of lines identical to line **264**. Because of the presence of the inclined, leading plane **270**, which forms an angle **269** with the velocity vector W_1, the gas flow is deflected clockwise by an angle equal to angle **269** (this angle is equal to the angle formed by the wedge-shaped leading portion of the blade), with the result that it will follow the line joining the points **263** and **261** at a reduced velocity but higher pressure than the pressure existing in front of the oblique shock wave front. If the relative velocity W_1 is sufficiently high, the oblique shock will have sufficient velocity and energy to be reflected from surface **252** to produce a reflected shock wave front **265**. The reflected shock again reduces the velocity of the flow stream and converts its kinetic energy into potential pressure energy so that the pressure on the inner side of the reflected shock wave front is higher than on its outer side. The maximum relative velocity W_1 and the angle **269** are proportioned to produce only two shocks **264** and **265**, with the result that all flow beyond shock **265** is subsonic, the reduced subsonic velocity vector W_2 existing throughout the remaining part of the channel. This remaining part, i.e., from point **261** to point **266**, is merely a turning channel, the degree of turning being determined by the magnitudes of W_2 and U_3, which is the peripheral speed of the second stage, and the minimum feasible angle of entry of W_2 into the compression channel of the second stage. The wave fronts **264** and **265** may or may not be straight planes, depending on the conditions of flow. Irrespective of the configuration of the shock fronts, the functioning of the compressor will be the same.

In order to make the subsonic portion of the flow channel merely a turning channel, its cross-sectional area, normal to the direction of flow should remain constant at all stages. Stated differently, area A_1 should be equal to area A_2, Fig. 16. To obtain this, it becomes necessary to reduce the axial width of the channels, as illustrated at **114** and **116** in Fig. 1. It is to be noted that the transverse dimension of the subsonic portion of the channel, as it appears in Fig. 2, is increasing very rapidly as one progresses inwardly toward the inner periphery **267** which is necessary to make the blades terminate in sharp trailing edges **256**. Thus, the channel, as it appears in Fig. 2, creates an impression as if it were a diffusing channel, and the correction illustrated in Fig. 1 is the correction which is necessary to transform it into a constant velocity channel. It is also to be noted that the above configuration increases the hydraulic radius of the channel toward the exit which is favorable for diminishing friction and the thickness of the boundary layer because of the continuous increase in the Reynolds' number. It is advantageous to maintain the velocity W_2 constant since greater compression ratio can be obtained in this manner in the second stage. This is so because the compression ratio of any stage is a function of the local Mach number to the second power.

The compression technique used in the second stage is identical to the technique used in the first stage, and therefore does not require a detailed description. Oblique and reflected shocks are used which create the respective compression wave fronts **275** and **276**. The main difference between the first and the second stages resides in the fact that the flow channel on the other side of the wave front **276** is a diffusion channel, the side walls **118** and **120**, Fig. 1, of the channel being parallel to each other, while the distance between the opposed surfaces of the blades **280** is increasing in centripetal direction. The rate of diffusion is determined by the use to which the compressed air is subjected. In the disclosed compressor, the compressed air is ducted into the combustion chamber 35, Fig. 1, which revolves at the same angular velocity as the second compression stage. It leaves the compressor at a relative velocity W_4 which is considered to be of proper magnitude to produce the desired degree of swirl in the combustion chamber, which in turn is determined by the rate of combustion desired in the combustion chamber.

Therefore, the degree of diffusion is a function of the rate of combustion. The relative velocity W_4 is, in this case, radial, and therefore, the abolute swirl velocity C_{u_4} is of the same magnitude as the peripheral velocity U_4 at this radius, i.e., $C_{u_4}=U_4$, and the absolute velocity C_4 forms a leading angle γ with the radial line. The shape of the blades in the second stage is determined in the supersonic region in the same manner as the shape of the blades of the first stage in the same region. The two differ in shape, however, because $W_1<W_3$. As to U_1 and U_3, U_1 may be made greater than U_3 if the angular velocities of the two rotors are made equal. However, it is more advantageous to make $U_1=U_3$. This will be discussed more in detail in connection with the vector diagram of the compressor illustrated in Fig. 7.

In Fig. 2, a compressor is disclosed which uses oblique and reflected shocks in both stages to obtain the compression of gases. This compressor obtains the highest compression ratio at the highest thermodynamic efficiency for the obtained compression ratio, and therefore, offers advantages over the compressors illustrated in Figs. 3, 4, and 5. However, the additional compressors illustrated in the above figures have some of the advantages of their own, as will be described later.

Referring to Fig. 3, it discloses a two-stage compressor having a stationary supersonic contra-prerotation stage and two supersonic compression stages which utilize only an oblique shock for obtaining compression. Since only oblique shock compression is used in the compression stages, the relative velocity W_1 need not be especially high, and therefore, the prerotation stage may be a completely subsonic stage. The shape of the blades 300 would then differ from the blades illustrated in Fig. 3 only in the lower part of the blades. While in Fig. 3 the flow channel terminates in the supersonic expansion nozzles of the types previously described with Fig. 2, the subsonic acceleration channel would then continue to the very end by gradually narrowing the dimension of the flow channel in the plane of the drawing, i.e., in the plane normal to the axis of rotation. A subsonic prerotation stage of this type is illustrated in Fig. 10. The fluid to be compressed leaves the prerotation stage at a subsonic or a supersonic absolute velocity C_1, and then enters the flow channel of the first stage at a relative supersonic velocity W_1, which must be a supersonic velocity irrespective of whether C_1 is subsonic or supersonic. The leading edges (on the pressure side) of blades 301, 302, etc., create oblique shocks, the wave fronts of which are illustrated by inclined lines 303, 304, etc. The configuration of the blades in this stage differs from the configuration of the blades illustrated in Fig. 2 by the omission of an edge 259 between flat surface 270 and curved surface 258 since no reflected shock is used in this compressor. Angle 311 must have the magnitude which will produce only an oblique shock of compression with the entry velocity W_1. The equation for this angle is given later in this specification, in connection with the discussion of Figs. 11 through 14. The remaining portion of the flow channel preferably is a constant velocity channel, as in the case of Fig. 2, since introduction of any diffusion in this channel would only produce the thickening of the boundary layer and reduce the exit velocity W_3, thus lowering the compression ratio obtainable in the second compression stage. Stated differently, greater compression ratio is obtained by maintaining W_3 as high as possible and therefore, any attempt of obtaining some compression by diffusion in the first stage would produce a disproportionate loss of compression in the second stage. The changes in the shape of the blades in the second stage, i.e., in the blades 304, 305, etc., are identical to the changes in the blades 301, 302, of the first stage, i.e., the edges 282, Fig. 2 now are eliminated since there is no reflected shock. The oblique shock wave fronts are illustrated by lines 306 and 307 in this figure. From then on, i.e., from point 308 to point 309, the channel is a subsonic diffusion channel where the compression obtained by the oblique shock 307 is increased subsonically to a still higher pressure by converting relatively high kinetic energy to potential energy through the process of subsonic diffusion. The advantages of this compressor, disclosed in Fig. 3, are several. Although it does not produce as high a compression ratio as that in Fig. 2, it nevertheless produces a somewhat lower compression ratio at the highest thermodynamic efficiency because of very nearly isentropic process of compression by the oblique shocks. It also will operate quite effectively and efficiently in the subsonic region, and therefore will "pull in" into and through the subsonic region without any difficulties. This is so because of the absence of sharp transverse edges 259 and 282, Fig. 2, which are apt to cause a certain amount of separation in the subsonic region, in the compressor disclosed in Fig. 2.

Fig. 4 discloses a compressor which utilizes oblique, reflected, and normal shocks. In this case, the relative velocity W_1 of the fluid at the exit from the prerotation stage must be high in order to produce three shocks in the compression stage. The prerotation stage, therefore, must have a supersonic nozzle, identical to the nozzle disclosed in Fig. 2, which will produce supersonic velocity W_1. Sides 400 and 401 of blades 402 and 403 are identical flat surfaces parallel to the relative velocity W_1. Angles 404 and 405 are computed according to the formula given in the latter part of the specification to produce oblique, reflected, and normal shocks with the given supersonic relative velocity W_1. The wave fronts of these shocks are 406, 407, and 408 respectively. One of the requirements of the compression channel utilizing normal shock is that its minimum width must be at the position where the appearance of the normal shock is desired, which is the case in Fig. 4. From then on, the channel may be either a diffusing channel or a constant velocity channel, the latter offering the advantages mentioned in connection with Fig. 2, i.e., higher compression ratio is obtained in the second stage when W_2 is maximum; since W_2 is maximum, W_3 is also maximum. The next stage is similar to the first stage up to, and including, the minimum channel width where normal shock 410 is produced. From then on the channel is a diffusing channel. The position of the median line 411 determines whether the absolute exit velocity C_4 is radial or has a rotational swirl, as may be desired in the disclosed power plant because the compressor is used with the rotating combustion heat generator.

The compressor in Fig. 4 produces the highest compression ratio because of the use of three shocks, but this high compression ratio is produced at a decreased thermodynamic efficiency. This compressor also presents some structural limitations in that the angles 404, 405, etc., are small and therefore the leading edges of the blades are less rigid. The "pulling-in" characteristic of the compressor disclosed in Fig. 4 is inferior to the pull-in characteristic of the compressor disclosed in Fig. 2, and even more inferior to the pull-in characteristic of the compressor disclosed in Fig. 3.

Fig. 5 discloses a compressor in which the prerotation stage has a supersonic nozzle, and the compression stages produce oblique and normal shocks. The compression ratio of this compressor can be as high as that of the compressor illustrated in Fig. 2 and also in Fig. 4 if the relative entry velocities, W_1, are equal in all cases.

All compression stages utilizing normal shock require sharp edges **409**, **414**, Fig. 4, **500** and **501**, Fig. 5, extending through the entire length of the channel, to locate and keep the position of the normal shocks in fixed positions with respect to the blades and at the minimum width of the channel. This will be discussed more fully in connection with Figs. 11 through 14.

Fig. 6 illustrates the method of reinforcing compression blades by means of thin vanes **600** and **601** placed midway across the blades **604** and **605** to prevent centrifugal deflection of the leading edges **602** and **603** of the blades. The leading edges **606** and **607** of the blades must be sharp edges, and these edges should be positioned behind the oblique shocks, where the Mach number is lower. These vanes are placed in slots, such as slot **608**, with which they form a tight fit. After their insertion into the slots, the vanes and the blades are brazed together. The vanes constitute integral parts of the blades. When the compressor blades are especially wide, several vanes may be used with each blade.

Fig. 7 illustrates the velocity vectors of all velocities present in the compressor of the type illustrated in Fig. 2. All of these velocities are also illustrated in Fig. 2 in proper relationship with respect to their actual location in the compressor stages. Therefore, the description of these vectors will be given in connection with Figs. 2 and 7. The fluid to be compressed enters the prerotation stage in radial direction with an absolute subsonic velocity C_0, this velocity being taken at the outer periphery **211**. In the prerotation stage, C_0 is transformed into an absolute supersonic velocity C_1 leaving the contra-prerotation stage at its inner periphery **201** at an angle of approximately 25° which is an optimum angle, and is determined by the angular position of the mean flow line **218** at the exit from the prerotation stage. Generally, from a theoretical point of view, this angle should be as small as possible, in order to produce as large a relative velocity W_1 as possible and large C_u. As it will be demonstrated in connection with Figs. 11 through 14 (Equation 5), the compression of any stage is a direct function of the product of the peripheral velocity, U, and the projection of the absolute velocities, C_1 and C_2, at the entry and exit of compression stage, upon the peripheral velocity U. Therefore, the smaller is the angle β_1, the greater is the stage compression. However, this angle cannot be made equal to zero because no fluid would enter the compression stage under such conditions, and there would be nothing to compress. Geometric solution of this problem involves composing the shapes of the blades of the prerotation stage, and more accurately stated, the shape of the flow channel of the prerotation stage and its proper angular relationship with respect to the entry channel of the first stage, the angle and the shape of the latter being dictated primarily by the peripheral velocity U_1 and β_1. The minimum value of β_1 is of the order of 15°. This angle is an inverse function of the radius of the compressor, and the above value is for the compressor in which the outer diameter of the first compressor stage is 32.5 inches. With the increase of the radius this angle decreases since there is a corresponding decrease in the curvature of all stages. The outer periphery of the first stage travels at the peripheral velocity U_1, the value of which should be as high as possible, and its limit is determined solely by the strength and weight of the available metals. For an ordinary steel it is of the order of 700 feet/second, and for titanium alloys it is of the order of 850 feet/second, maximum. For vacuum-processed steels it may be as high as 1000 feet/second. The relative velocity, W_1, then merely closes the velocity triangle. As mentioned before, β_1 is then the angle between W_1 and the tangent line **900** drawn through the point or origin **901** of the vectors. This point lies on the outer periphery of the first stage. The absolute radial velocity of the fluid is C_r. The significance and the limits of the Mach numbers will be discussed in connection with Figs. 8 and 9.

The vector diagram for the second stage is derived in the same manner. The fluid leaves the first stage with relative velocity W_2 and absolute velocity C_2. The first stage has a peripheral velocity U_2 at its inner periphery **267**, Fig. 2. At this stage, it encounters the oppositely rotating second stage, the outer periphery of which rotates at the peripheral velocity U_3, which is opposite to U_2 because of the contra-rotation of the second stage with respect to the first stage. This produces a relative velocity W_3 at which the fluid enters the second stage. The radial velocity C_{r_2} is increased and, therefore, $C_{r_2}>C_{r_1}$. Shock compression and diffusion in the second stage reduce all velocities, the degree of the final reduction being determined by the degree of diffusion in the diffusion channel, which may be made greater or smaller, according to the final use of the compressed gas. As mentioned previously, the exit absolute velocity C_4 may be radial or follow the direction of rotation, as illustrated by the two triangles in Fig. 9, its direction being determined by the angle the mean flow line **279** makes with the inner periphery **285** of the second stage.

Optimum operating conditions for the supersonic centripetal compressor

Fig. 8 illustrates a temperature-entropy diagram of a compression cycle of a dynamic compressor with a prerotation stage. This diagram will be discussed primarily in connection with the performance of the centripetal compressor. However, it is equally applicable to all other types of compressors, axial and centrifugal, if they are provided with stationary prerotation stages.

The atmospheric pressure, P_0, is the pressure at which the air enters the prerotation stage; the entry velocity is C_0, and is equal to zero, which means that the ambient air is at a standstill. This point is indicated as point **0** on the diagram. The first stage of the compressor is the stage which accelerates this ambient static air from velocity C_0 to velocity C_1, which may be either supersonic or subsonic velocity, depending upon the pressure differential created by the first and second compressor stages.

In Fig. 8, this velocity, according to Bernoulli's theorem, produces a corresponding pressure drop to P_1, which is lower than the atmospheric pressure, this loss in pressure corresponding to the gain in the kinetic energy of the air entering the compressor. This loss in pressure head is expressed by

$$\Delta H_{C_1}=\frac{C_1^2}{2g}\ \frac{\text{ft. lbs.}}{\text{lb.}} \qquad (2)$$

The higher is the exit velocity C_1 from the prerotation stage, the larger is the value of ΔH_{C_1}, and it becomes necessary to determine what is the optimum value of C_1 for obtaining maximum compression ratio. Conditions may arise when this loss in pressure may be so high that the recovery of this pressure in the first compression stage either would not be possible altogether, or only a minor recovery would be obtainable, in which case the advisability of using the prerotation stage, at first glance, may be questioned altogether. As will appear later, the presence of the prerotation stage is more than justified under all circumstances, and that the only parameter which requires a closer scrutiny is the value of C_1 which controls, at least insofar as the practical operating range is concerned, the pressure ratio of the entire compressor. Thus, this velocity determines whether the prerotation stage should be subsonic or supersonic, and whether the first compression stage utilizes only an oblique shock or oblique-and-reflected shocks for compressing the air.

The local Mach number of velocity C_1 is expressed by

$$M_{c_1}=\frac{C_1}{a_{c_1}}=\frac{C_1}{a_0^2-\left\{\frac{k-1}{2}\right\}C_1^2} \qquad (3)$$

where

M_{c1}=local Mach number of C_1
a_{c1}=local sonic velocity, at point 1, Fig. 8
a_0=sonic velocity at point 0, Fig. 8
k=ratio of specific heats,

$$\frac{C_p}{C_v}$$

The air, after being accelerated to velocity C_1, enters the first compression stage, which accelerates this air to higher absolute exit velocity C_2 and at the same time compresses it to pressure p_2, this state of the air being illustrated by point 2 in the diagram. The rise in pressure is denoted by the positions of the respective points on the constant pressure lines P_0 through p_8. The pressure gain obtained in the first compression stage corresponds to the pressure difference between p_2 and p_1. This pressure gain is used for deriving a pressure head, ΔH_{p1} and mechanical head ΔL_1 from the following equations:

$$\Delta H_{p_1}=\frac{1}{g}\left\{C_{u_1}U_1-C_{u_2}U_2-\frac{C_2^2-C_1^2}{2}\right\}\frac{\text{ft. lbs.}}{\text{lb.}} \quad (4)$$

and

$$\Delta L_1=\frac{1}{g}\{C_{u_1}U_1-C_{u_2}U_2\}\text{ ft. lb./lb.} \quad (5)$$

where

ΔH_{p1}=pressure head obtainable in the first compression stage
g=acceleration due to gravity
C_{u1}=projection of C_1 on U_1
U_1=peripheral velocity of the outer periphery of the first stage
C_{u2}=projection of C_2 on U_2
U_2=peripheral velocity of the inner periphery of the first stage
C_2=absolute exit velocity of air at the exit from the first stage
C_1=entry velocity into the first stage and exit velocity from the prerotation stage.

Also, ΔH_{p0}, which is the net pressure head gain in the first stage, i.e., actual pressure gain above static atmospheric pressure, is derived from the equation

$$\Delta H_{p0}=\frac{1}{g}\left\{C_{u_1}U_1-C_{u_2}U_2-\frac{C_2^2}{2}\right\} \quad (6)$$

where the terms are the same as those in the Equations 2 through 5.

Besides producing the above pressure head gain, the first stage also increases the kinetic energy head by the amount ΔH_k, which is

$$\Delta H_k=\frac{C_2^2-C_1^2}{2g} \quad (7)$$

and, therefore, the total head rise in the first stage above static atmospheric stage, which is represented by the vertical distance between points 0 and 3, and designated by H_0 in Fig. 24, is

$$\Delta H_0=\Delta H_{p0}+\Delta H_k=\Delta L_1-\Delta H_{C1} \quad (8)$$

The most important quantities are ΔH_0, ΔH_{p0} and ΔL since they determine the net performance of the first stage. The quantities are plotted against Mach number of the velocity C_1, in Fig. 9, which will be discussed more in detail later.

The compressed air leaves the first compression stage with an absolute velocity C_2 which is larger than C_1, the velocity appearing in Fig. 8 is the velocity head ΔH_{C2} which is equal to

$$\Delta H_{C_2}=\frac{C_2^2}{2g} \quad (9)$$

Therefore, point 4 corresponds to the stagnation pressure of velocity C_2, the corresponding pressure line being p_s. The distance between the points 2 and 3 corresponds to the velocity head increase H_k due to the mechanical head supplied by the power delivered to the first compression stage. It is equal to

$$\Delta H_k=\frac{C_2^2-C_1^2}{2g} \quad (7)$$

It is to be noted here that the absolute exit velocity C_2 should be larger than C_1 for optimum overall performance, which will produce a large compression ratio in the second stage. That C_2 should be larger than C_1 follows from the velocity diagram in Fig. 7. This velocity, $C_2 \equiv C_3$ is a function of β_3 and of the peripheral velocity U_3 of the succeeding stage and optimum compression ratio of the entire compressor is obtained when C_2 is as high as possible and larger than C_1. As stated previously, $C_2 \equiv C_3$ is maximum when β_3 is of the order of 15°. It may be also stated here that C_3 may be made smaller than C_1, but this can be done only at the expense of the overall compression ratio.

The compression cycle in the second stage begins at point 2 rather than point 4, since point 4 represents the stagnation condition of stage 1. The mechanical head of the second stage, ΔL_2, is supplied by the mechanical energy delivered to the second compressor stage by the first stage of the turbine. The pressure head of the second stage, ΔH_{p2}, is from point 2 to point 6, and subsequent subsonic diffusion produces an additional gain in pressure head, ΔH_d, which is

$$\Delta H_d=\frac{C_7^2-C_6^2}{2g} \quad (10)$$

thus raising the pressure head from point 6 to point 7. The absolute exit velocity, C_7, whose Mach number is of the order of 0.2, is represented in the diagram as the velocity head

$$\Delta H_{c_7}=\frac{C_7^2}{2g} \quad (11)$$

thereby reaching stagnation pressure p_8 at point 8. The pressure ratio p_8/p_0 is the total compression ratio of the compressor.

The above head-entropy diagram is for the two stage compressor. For the three stage compressor, the diagram should be continued in similar manner for one additional stage.

Before concluding the description of Fig. 8, it is noteworthy to point out the important role being played by the prerotation stage in the compressor. The prerotation stage increases the velocity C_1 and its peripheral component C_{u1} as indicated in Fig. 7. Since the mechanical head ΔL_1 is equal to

$$\Delta L_1=\frac{1}{g}(C_{u_1}U_1-C_{u_2}U_2) \quad (5)$$

it is obvious that ΔL_1 is a function of C_{u1}. The effect of C_1 on ΔH_0 is illustrated in Fig. 9, and extrapolation of this curve to $M_{c1}=0$, when ΔH_0 is also equal to zero, indicates that there is a very rapid decrease in ΔH_0 with the decrease in C_1. This dotted region corresponds to the region of operation when the compressor does not have any prerotation stage. Low compression ratio in the first stage obviously will influence the remaining stages as well.

Proceeding now with the description of Fig. 9, examination of the ΔH_{p0} and ΔL_1 curves discloses that these heads increase continuously with the increase of M_{c1} or with the increase of C_1. The curve for ΔH_0, however, reaches its maximum at $M_{c1} \approx 0.9$, and then decreases to a constant value when $M_{c1} \approx 2.4$ or 2.5. Fig. 9 also includes two additional curves, one for ΔH_{c2} and the other for ΔL_T. ΔH_{c2} is the velocity head of the absolute exit velocity from the first stage, and is equal to

$$\Delta H_{c2}=\frac{C_2^2}{2g} \tag{9}$$

while ΔL_T is the total mechanical head of the compressor, and is equal to

$$\Delta L_T=\frac{1}{g}\{C_{u_1}U_1+C_{u_2}(U_2+U_3)+C_{u_4}U_4\} \tag{12}$$

where

U_3=peripheral velocity of the outer periphery of the second stage

C_{u_4}=projection of the absolute exit velocity, C_4, from the second stage on U_4

U_4=peripheral velocity of the inner periphery of the second stage

In Equation (12), the directional signs of the velocity vectors have been taken into consideration, and therefore, it is applicable only to the vector diagrams as they are used here.

Examination of the curve for ΔL_T discloses that ΔL_T increases slowly with the increase of M_{c_1} or C_1. Therefore, it would appear that M_{c_1} should be made at high as possible for obtaining maximum overall compression. This, however, is not so for two reasons. One reason is the increase in friction losses at high velocities, which do not appear in Fig. 9. Another reason is of purely mechanical nature: the shape of the leading edges of the cascades assume the form of wedges having progressively larger includes angle (angle **269** in Fig. 2) with the result that the concave surface forms a progressively smaller angle with the inner side of the wedge. This will produce separation at subsonic velocities during the starting period, which will create starting difficulties . Therefore, the maximum limit for M_{c_1} is somewhere between 1.5 to 2.0. Line **900** indicates the point for the two-stage compressor illustrated in Fig. 10.

If the maximum limit for M_{c_1} is in the region indicated above, there is also a minimum limit which is in the region of $M_{c_1}\approx.7$ to .9, although the compressor will produce compression even in the region lower than $M_{c_1}\approx.7$, but the second stage will need a large subsonic diffuser to convert very high kinetic energy into pressure. The above minimum values for M_{c_1} definitely indicate that the prerotation stage may well be a subsonic stage. This will require blades in both compressor stages which would produce oblique shocks only, which is especially the case for $M_{c_1}\approx.7$, since the kinetic energy of air will be insufficient to produce an oblique as well as the reflected shock. This type of compressor has distinct advantages in that, although the overall compression ratio may be somewhat lower, its operating characteristics become much more flexible in terms of rotational speeds. For example, such compressor will pull in during starting periods with greater ease and lower rotational speed because of the shape of the blades which produce flow channels substantially devoid of separation dangers at lower speeds.

An additional combination of stages is also possible. If the prerotation stage is subsonic, the first stage in all likelihood will be only an oblique shock stage because of low velocity C_1. However, since there is only a limited reduction in the air velocity through the first stage when there is only an oblique shock, and what is more important, the two contra-rotating stages produce the summation of U_2 and U_3, the resulting relative velocity W_3 becomes a very large supersonic velocity. Therefore, the second compression stage may be of the type in which the oblique as well as the reflected shock become possible. This combination of the stages is disclosed in Fig. 10.

The above demonstrates that different types of stages may be combined with each other to produce the sought results, from which the following table results.

SUPERSONIC OR SUBSONIC PREROTATION STAGE

1st Compression Stage	O.R	O	ORN	OR	O	ORN	ORN	O	OR
2nd Compression Stage	O.R	O	ORN	O	OR	OR	O	ORN	ORN

where

O=oblique shock stage

OR=oblique and reflected shock stage

ORN=oblique, reflected, normal shock stage

No stage is suggested which utilizes only the normal shock since the efficiency of such compressor would be quite low in comparison to the suggested combinations. It also has mechanical limitations due to the difficulty of anchoring this shock at the flat surface of adjacent blade, although this type of compression is possible in connection with the centripetal type of compression.

Supersonic compression by means of supersonic compression shocks

Although it has been known that the supersonic compression shocks constitute effectime means for obtaining efficient compression of gases within extremely limited space, the subject invention constitutes the first application of the supersonic compression shocks to centripetal compressors for obtaining compression of gases at high efficiency.

The disclosed methods of supersonic compression are based upon the principle of inducing and localizing the geometrical position of the shocks to the input side of the flow channels. Stated differently, the blade structure is such that there is a highly selective production of the desired shocks and elimination of all parasitic shocks; the elimination of the parasitic shocks is imperative for obtaining supersonic compression of gases, and the disclosed method of compression is impossible without such elimination. Therefore, the disclosed channel structures are adapted not only to produce supersonic compression shocks in proper geometric relationship with respect to the geometry of the flow channels in the centripetal compressors, but they are also constructed to eliminate the parasitic shocks in adjacent channels and in the gap between the stages. This will appear more clearly in the course of the discussion which is to follow.

Fig. 15 discloses the cross-sectional view of a two-dimensional, leading edge of a blade in a free supersonic stream of flow at supersonic velocity W_1. This leading edge **1500** is formed by the intersection of two flat surfaces **1501** and **1502**, which thus form a wedge-shaped structure **1503**. It is to be noted at once that one of the flat surfaces, i.e., surface **1501**, which is the lagging, or the suction, part of the blade in the disclosed compressors, is parallel to the velocity W_1 and therefore, apart from the frictional effects, is incapable of producing any disturbance in the stream, i.e., it does not produce any supersonic compression shocks which would otherwise interfere with the desired shocks.

The other surface **1502**, which is the leading side, is inclined at an angle δ with respect to the direction of flow and, therefore, forces the flow to deflect from the direction of vector W_1 to the direction of vector W_2, the latter being parallel to the wedge wall **1502**. Such deflection produces an oblique planar shock **1504** which separates the disturbed region of flow from the undisturbed region. The principal variables on two sides of the shock are M_1, T_1, p_1, ρ_1, W_1 on one side and M_2, T_2, p_2, ρ_2, W_2 on the other side. They will be discussed more fully in connection with the equations which follow.

The physical change of gas passing through the shock must satisfy the equation of continuity, which is

$$\rho_1 W_{n_1} = \rho_2 W_{n_2} \tag{14}$$

where

ρ_1=gas density before the shock
ρ_2=gas density after the shock;
W_{n_1}=normal velocity component of W_1; which forms 90° angle with the plane of the shock (see Fig. 15); it is a function of W_1;
W_{n_2}=normal velocity component of W_2; which forms 90° angle with the plane of the shock; (see Fig. 15).

The momentum equation must be satisfied also; it is

$$p_1 + \rho_1 W_{n_1}^2 = p_2 + \rho_2 W_{n_2}^2 \tag{15}$$

and, from 15

$$\rho_1 W_{n_1} W_{t_1} = \rho_2 W_{n_2} W_{t_2} \tag{16}$$

where

p_1=absolute pressure ahead of the shock,
p_2=absolute pressure after the shock;
W_{t_1}=component of the stream velocity W_1 in the plane of the shock (see Fig. 15)
W_{t_2}=component of the stream velocity W_2 in the plane of the shock.

The third equation which must be satisfied is the equation of conservation of energy which is

$$\frac{K}{(K-1)}\cdot\frac{p_1}{\rho_1} + \frac{1}{2}W_1^2 = \frac{K}{(K-1)}\,\frac{p_2}{\rho_2} + \frac{1}{2}W_2^2 \tag{17}$$

where K is the ratio of specific heats.

From (14) and (16) it follows that:

$$W_{t_1} = W_{t_2} \tag{18}$$

and from Fig. 15

$$\frac{W_2}{W_1} = \frac{\cos\delta}{\cos\alpha} \tag{19}$$

Using

$$a_1^2 = K\frac{p_1}{\rho_1}$$

where a is the sonic velocity before the shock, it follows

$$\rho_1^2 W_{n_1}^2 = \rho_1^2 W_1^2 \sin\vartheta = \rho_1 M_1^2 K p_1 \sin^2\vartheta \tag{20}$$

From the above equation, it is possible to derive the fundamental equations of shocks, which are:

$$\frac{p_2}{p_1} = \left(\frac{T_2}{T_1}\right)^{\frac{K}{K-1}} = \frac{2K}{K+1}\left(M_1^2 \sin\vartheta - \frac{K-1}{2K}\right) \tag{21}$$

$$\frac{\rho_1}{\rho_2} = \frac{2}{K+1}\left(\frac{1}{M_1^2 \sin^2\vartheta} + \frac{K-1}{2}\right) \tag{22}$$

and from (16) and (18)

$$\frac{\rho_1}{\rho_2} = \frac{W_{n_2}}{W_{n_1}} = \frac{W_{n_2}\cdot W_{t_1}}{W_{t_2} W_{n_1}} = \frac{\tan\alpha}{\tan\vartheta} \tag{23}$$

therefore,

$$\frac{\tan(\vartheta-\delta)}{\tan\vartheta} = \frac{2}{K+1}\left(\frac{1}{M_1^2 \sin^2\vartheta} + \frac{K-1}{2}\right) \tag{24}$$

Equations (23) and (24) are general equations and are applicable to all compression shocks (oblique, reflected, normal). For normal shocks

$$\vartheta = \frac{\pi}{2} \text{ and } \delta = 0$$

the flow, therefore does not change direction and the shock is normal to the flow. Therefore, it is called the normal shock. The normal shock gives the maximum obtainable pressure ratio for a given Mach number in front of the shock.

For the normal shock, from 21

$$\left(\frac{p_2}{p_1}\right)_n = \frac{K(2M_1^2-1)+1}{K+1} \tag{25}$$

where subscript n means that the pressure ratio is for the normal shock and

$$\left(\frac{\rho_1}{\rho_2}\right)_n = \frac{M_1^2(K-1)+2}{M_1^2(K+1)} \tag{26}$$

and for the Mach number behind the normal shock

$$M_2^2 = \frac{M_1^2 + \frac{2}{K-1}}{\frac{2K}{K-1}M_1^2 - 1} \tag{27}$$

Figs. 11 through 14 illustrate the application of the oblique, reflected and normal shocks in four possible combinations, which are

(1) Oblique shock only (Fig. 11).
(2) Oblique in combination with normal shock (Fig. 12).
(3) Oblique and reflected shocks (Fig. 13).
(4) Oblique, reflected and normal shocks in combination (Fig. 14).

In Fig. 11, one of the important features is that surface **1100** is parallel to W_1. This prevents generation of any parasitic shocks, which would interfere with the entire phenomena of compression. δ has the same significance as δ in Fig. 15. Its value can be derived from Equation (24); this value is not critical. Meeting of the flat surface **1100** with a cylindrical convex surface **1101** establishes an edge **1102**. The position of this edge is chosen so that the oblique shock terminates at this edge. From then on, the channel may be a constant velocity subsonic channel or a subsonic diffusion channel. In either case, it may be either a straight or a curved channel, depending on the desired direction of W_2.

In Fig. 12, δ is so chosen that W_2 is still supersonic, and therefore, the lower flat surface **1200** of the wedge is parallel to W_2. When the fluid reaches the narrowest portion of the channel, which is indicated by points **1201** and **1202**, a normal shock appears at this portion, changing the flow to velocity W_3, which is subsonic.

It may be shown that the relationship between M_2 and ϑ is expressed by the equation

$$\left\{\frac{1}{M_1^2\sin\vartheta} + \frac{K-1}{2}\right\}\left\{\frac{1}{M_2^2\sin(\vartheta-\delta)} + \frac{K-1}{2}\right\} = \left(\frac{K+1}{2}\right)^2 \tag{28}$$

The above equation is solved for ϑ after making M_2 larger than 1 for the condition illustrated in Fig. 12.

The shocks illustrated in Fig. 13 are oblique and reflected. The reflected shock may be considered as no more than the second oblique shock, and therefore, what has been said of the oblique shock illustrated in Fig. 11 also applies to the reflected shock illustrated in Fig. 13. The reflected shock is anchored at edge **1300**. From then on, the channel is either a subsonic diffusion or a constant velocity subsonic channel.

Fig. 14 merely combines everything that has been said before of all previous channels, since in this case all three shocks are present. W_1 should be higher here than in any previous case to justify the use of such complex shock pattern.

The invention thus discloses the novel methods of compressing fluid in a centripetal supersonic compressor in which the fluid is first accelerated to a supersonic or subsonic absolute velocity C_1 in the stationary contra-prerotation stage. The direction of velocity C_1 is such that it tends to be in opposition to the direction of the peripheral velocity U_1 of the first compression stage. Obviously, there cannot be an angle of 180° between C_1 and U_1, and this angle in practice is of the order of 160°. However, what is meant by the expression that C_1 tends to be in opposition to U_1, is that U_1 points almost in the opposite direction to C_1; i.e., that the contra-prerotation stage contra-prerotates the fluid with respect to U_1. This increases W_1 still further, which enables one to obtain very high relative velocities (Mach number of W_1 of the order of 2.3) although the peripheral velocity of the compression stages is not especially high, of the order of 830 feet per second. The same principle is used between the succeeding stages, which are contra-rotating with the result that W_3 is obtained by the vectorial addition of U_2 and U_3 (see Fig. 7, and especially triangle W_3–W_2–U_3–U_2), which obviously increases the value of W_3 approximately 2.5 times as compared to W_2, i.e., 250%. This obviously produces much more intense shocks than those which would have been obtainable with C_1. It is to be noted also that the supersonic compression shocks are produced only on one side of the leading edges of the blades (see Figs. 11 through 14, for example), which is accomplished by making one plane surface, such as surface **1100** in Fig. 11, parallel to the direction of flow of the fluid with the result that this plane surface does not produce any compression shock, and only the opposite surface of the wedge, such as plane surface **1200** in Fig. 12, is made to produce the compression shocks by being inclined at an angle δ to the direction of flow.

The disclosed methods also indicate that a single or a plurality of different shocks may be used for obtaining the desired degree of compression.

The specification refers to stator **69** (Fig. 1) as a "stationary contra-prerotation stage." This term, as used in the specification and claims, means a stationary device inducing an absolute exit velocity C_1 which has a radial, centripetal flow component C_{r_1} and a peripheral component C_{u_1} (see Fig. 7). C_{u_1} has the direction which is directly opposite to the direction of the peripheral velocity U_1, and it is for this reason that stator **69** is referred to as the "contra-prerotation" stator, i.e., it is the stator which produces velocity component C_{u_1}, contra to U_1. U_1 represents the rotation component, and, therefore, C_{u_1} is contra to the direction of rotation.

It should be understood also that the flow of the fluid being compressed, obviously, takes place only when an external power is furnished to the two compressor shafts **31** and **43**. This is discussed at great length throughout the preceding portion of the specification

The claims also refer to the "centripetal flow" stages. Such flow need not be a purely radially directed centripetal flow, such as C_{r1}, but also includes such flows as C_0, C_1, W_1, C_2, W_2, W_3, etc., which may (C_1, W_1, etc.) or may not (C_0, W_4, C_{r_1}, C_{r_2}) include a peripheral component, but all of which include the centripetally-directed purely radial component.

What is claimed as new is:

1. A centripetal compressor for dynamically compressing a working fluid, said compressor comprising a stationary, centripetal flow contra-prerotation stage having two side-walls and a plurality of cambered blades mounted between said side-walls; said blades being shaped to define acceleration channels for accelerating said fluid, at the exit from said stationary stage, to a maximum absolute velocity of said fluid within said compressor when said compressor is in operation; and a plurality of centripetal flow compression stages, each stage having a plurality of cambered blades defining respective centripetal flow channels; said blades having wedge-shaped leading portions, formed by first and second flat surfaces, for compressing said fluid by means of supersonic shocks when said compressor is in operation; said first surface being a leading surface and said second surface being a lagging surface of said blades, said lagging second surface being parallel to the direction of flow of said fluid at the entry of said fluid into the respective compression stage, said direction of flow being represented by the supersonic relative velocity of said fluid at the entry into the respective compression stages when said compression stages are rotated at their normal, operating angular velocities, said leading surface creating at least an oblique supersonic compression shock in said fluid, said shock extending diagonally into the flow channel from the tip of said wedge-shaped leading portion to the inner portion of said second flat, lagging surface.

2. A centripetal compressor for dynamically compressing a fluid, said compressor comprising a stationary centripetal flow contra-prerotation stage having cambered blades positioned between two side-walls, the adjacent blades and said side-walls forming, or defining a plurality of flow channels, each channel including a first portion for accelerating said fluid to a centripetally directed subsonic velocity and a second portion, following said first portion, for accelerating said fluid to a centripetally directed supersonic velocity, and at least two concentrically mounted contra-rotatable supersonic compression stages, each compression stage having a plurality of cambered blades having wedge-shaped leading portions, each wedge-shaped portion including a first, leading flat surface and a second, lagging flat surface. said first surface forming a sharp angle with the relative velocity of said fluid at the entry into the respective compression stages, for producing supersonic compression shocks in said fluid when said compression stages are rotated at their normal, operating, angular velocities.

3. The centripetal compressor as defined in claim 2 in which said second surface makes a sharp angle with said first surface, and a third cylindrical surface meeting said second surface and forming an obtuse angle with said second surface, said first surface producing an oblique compression shock and said second surface producing a reflected compression shock.

4. A centripetal compressor for dynamically compressing a fluid with the aid of supersonic shocks created in said fluid, said compressor including a stationary centripetal flow contra-prerotation stage having two opposing side-walls and a plurality of blades uniformly distributed around the periphery of said side-walls, the opposing surfaces of said blades and of said side-walls defining a plurality of centripetal flow channels, said flow channels having a subsonic acceleration portion for said fluid beginning at the outer periphery of said stage, said subsonic acceleration portion blending into a centripetal flow supersonic acceleration region for said fluid terminating at the inner periphery of said stage, said supersonic region being defined by a plurality of supersonic nozzles uniformly distributed around the periphery of said stationary stage for discharging said fluid at an absolute velocity C_1, said nozzles being formed by the opposing surfaces of said blades adjacent to the inner periphery of said stage, and a centripetal flow compression stage rotatable with a peripheral velocity U_1, said velocity C_1 having a purely radial component C_{r_1} and a peripheral component C_{u_1} having the opposite direction to that of U_1.

5. A centripetal compressor for compressing a fluid, said compressor including a stationary centripetal flow contra-prerotation stage followed by a centripetal flow compression stage receiving a fluid at a supersonic relative velocity W_1 when said compression stage is rotated at its normal, operating angular velocity, said compression stage including a plurality of blades having wedge-shaped leading edges producing oblique shocks in said fluid, each of said edges being formed by an intersection of a first flat leading surface and a second flat lagging surface, said second lagging surface being parallel to the direction of the relative velocity W_1, and said second surface being parallel to W_2, where W_2 is the velocity of said

fluid upon emergence from said shock, the two surfaces forming a sharp angle, the magnitude of said angle being a function of the relative local Mach number M_{W_n} at the entry into said compression stage.

6. A centripetal compressor for compressing a fluid with the aid of supersonic acoustic shocks, said compressor including a stationary centripetal flow contra-prerotation stage followed by a centripetal flow compression stage capable of receiving said fluid at a supersonic relative velocity W_1, said stage including a plurality of blades, the opposing surfaces of said blades defining flow channels, each blade having a leading edge formed by the intersection of a first leading plane surface and a second lagging plane surface, the second, or lagging, surface being parallel to the direction of said relative velocity W_1, and the first surface being at an angle δ with respect to the first surface, said first leading surface, upon encountering said fluid, producing an oblique supersonic compression shock in said fluid, said shock extending, from the edge of one blade to the inner portion of the second surface of adjacent, upstream blade, said angle δ having any value capable of producing said oblique shock, and being a function of the relative local Mach number.

7. A supersonic centripetal compressor having at least one centripetal flow compression stage, said stage comprising two side-walls, a plurality of blades mounted between said side-walls and spaced from each other and proportioned to receive a compressible fluid at a relative supersonic velocity W_1, the opposed surfaces of each pair of adjacent blades and said side-walls forming four boundaries of a flow channel, each channel beginning with a first flat surface, on a corresponding blade, parallel to said relative velocity W_1, said first surface merging with a second convex surface, the junction line between said first and second surfaces producing an edge normal to the direction of flow of said fluid through said channel and extending across the depth, or width, of said channel, and the opposed surface of said channel beginning with a leading edge produced by the junction of a third, leading flat surface on an adjacent blade with a surface on said adjacent blade which is parallel to W_1, said leading third flat surface being inclined to said velocity W_1 for producing an oblique shock in said fluid when said fluid enters said compression stage at said relative velocity W_1, said oblique shock terminating at substantially said transverse edge.

8. The supersonic centripetal compressor as defined in claim 7, in which the remaining portion of said channel is defined by said convex surface on one side, and a concave surface on the opposing side, said convex and concave surfaces, in combination with the two side-wall surfaces, completing the boundaries of said channel, said remaining portion of the channel having a cross-sectional area for producing constant velocity channel.

9. A centripetal compressor for dynamically compressing said fluid, said compressor comprising a stationary centripetal flow contra-prerotation stage at the entry of said fluid into said compressor, said stage having means for centripetally accelerating the flow of said fluid through said stage, and at least two contra-rotatable centripetally compressing compression stages, each compression stage having a plurality of cambered blades, each blade having a leading flat surface and a lagging flat surface to produce a wedge-shaped leading portion of the blade, the leading flat surface producing at least an oblique supersonic compression shock in said fluid upon its entry into the respective compression stage when said compressor is in operation.

10. A centripetal compressor as defined in claim 9, in which said compression stages also include the additional means for producing a normal supersonic compression shock in said fluid.

11. A centripetal compressor for dynamically compressing a fluid, said compressor comprising a stationary centripetal flow contra-prerotation stage having a plurality of cambered blades uniformly distributed around the periphery of said stage, said blades being mounted between two side-walls, said blades and said side-walls defining a plurality of centripetal flow channels for accelerating said fluid, at least first and second concentrically mounted supersonic compression stages, means for rotating said first stage in a clockwise direction and said second stage in a counter-clockwise direction at angular velocities having a fixed ratio with respect to each other, and a plurality of cambered blades and side-walls for supporting said blades in each compression stage, said last-recited blades having wedge-shaped leading portions formed by leading and lagging flat surfaces, said leading surface producing an oblique supersonic shock in said fluid, and a ridge portion at the downstream end of said flat lagging surface for anchoring a reflected supersonic shock produced by said lagging surface.

12. A dynamic compressor for compressing a fluid in a centripetal direction with the aid of supersonic acoustic shocks, said compressor comprising a stationary centripetal flow contra-prerotation stage for accelerating said fluid in a centripetal direction, and at least two centripetal flow compression stages rotatable in two opposite directions, each stage having a plurality of cambered blades having wedge-shaped leading portions formed by first and second intersecting surfaces, said first surface being a leading surface and said second surface being a lagging surface, the second surface being parallel to the direction of flow of said fluid along said second surface, and said first surface forming a sharp angle with said second surface for producing an oblique shock in said fluid, and a third concave surface meeting said first surface and forming an obtuse angle with said first surface and also forming a ridge at the line of intersection with said first surface for anchoring at said ridge a reflected shock produced by the second surface in said fluid.

13. A centripetal compressor comprising a first rotor including a first side-disc and a first hoop-ring connected to said first side-disc, a first set of blades, one end of said first set of blades being connected to said first hoop-ring, a first hollow cylinder having first and second ends, said first end being connected to the other end of said first set of blades, said first hollow cylinder being connected to a source of power for rotating said first rotor, first means rotatively supporting said first side-disc; and a second rotor including a second side-disc adjacent to said first side-disc and a second hoop-ring connected to said second side-disc, a second set of blades, one end of said second set of blades being connected to said second hoop-ring, a second hollow cylinder connected to the other end of said second set of blades, said second hollow cylinder being connected to a source of power for rotating said second rotor, and second means rotatively supporting said second side-disc, said first and second rotors being in concentric relationship with respect to each other.

14. A centripetal compressor comprising at least first and second hollow cylinders including, respectively, first and second centripetal flow compression stages integrated, respectively, into said cylinders, and first through fourth side-discs and first through fourth bearing means for supporting said cylinders and said side-discs in rotative and concentric relationship with respect to each other.

15. A centripetal compressor for dynamically compressing a fluid, said compressor including a first, or outer, centripetal flow compression stage having a plurality of centripetal compression channels, each channel having a converging upstream portion terminating in a throat, said upstream portion being defined by two sidewalls and first and second flat transverse walls both inclined in the direction of rotation of said first compression stage and also converging toward each other; the downstream ends of said first and second walls terminating at and defining said throat; said first wall being the lagging wall and said second wall being the leading

wall when viewed in the direction of rotation of said compression stage; a stationary centripetal flow contra-pre-rotation stage surrounding the outer periphery of said first compression stage; said stationary stage having a plurality of flow-accelerating channels having downstream portions turned, or cambered, in the direction opposite to the direction of rotation of said first compression stage; the degree of the inclination of the respective first and second transverse walls and the degree of turning of the flow-accelerating channels in said stationary stage being proportioned to discharge said fluid from the stationary stage and into the first compression stage at a supersonic relative velocity parallel to the leading transverse wall of said compression channel and at an angle to the lagging transverse wall of the compression channel when said compression stage is rotated at its normal, operating, peripheral velocity, said lagging transverse wall producing at least an oblique shock in said fluid upon its entry into said first compression stage.

16. The compressor as defined in claim 15 in which the downstream end of said second wall merges into a convex transverse wall, along a first transverse line, and the downstream end of said first wall merges into a concave transverse wall, along a second transverse line, the two transverse lines defining the position of said throat, said throat producing a normal shock in said fluid.

17. The compressor as defined in claim 15 which also includes a transverse, leading convex wall making a first junction with the downstream end of the first, or leading, wall; and a transverse, lagging concave wall making a second junction with the downstream end of the second, or lagging, wall; said oblique shock extending from the upstream edge of said lagging flat wall to said first junction, said first junction producing a reflected shock in said fluid, said reflected shock following said oblique shock and extending from said first junction to said second junction.

18. The compressor as defined in claim 15 which also includes a transverse, leading convex wall making a first junction with the downstream end of said flat leading wall, and a transverse lagging concave wall making a second junction with the downstream end of said flat lagging wall, said two junctions and adjoining surfaces of the two side-walls determining and defining the throat portion of said channel, said oblique shock extending from the upstream edge of said lagging flat wall to the downstream portion of said leading flat wall, said last-mentioned wall producing a reflected shock following said oblique shock and terminating at the second junction, said reflected shock being followed by a normal shock located at the throat of said compression channel.

19. The compressor as defined in claim 15 which also includes a second compression stage following said first stage, said second stage having a plurality of compression channels possessing the same compression means as the compression means of the channels of the first stage, and means for rotating said first and second compression stages at two opposite angular velocities having a fixed velocity ratio with respect to each other.

20. The compressor as defined in claim 15 in which said compression channel, on the downstream side of said throat, has a constant cross-sectional area for conveying said fluid at constant subsonic speed through the downstream portion of said compression channel.

21. The compressor as defined in claim 15 in which the compression channels of the first compression stage also include downstream portions each defined by said side-walls and a first, leading, convex, transverse wall on one side and a second, lagging, concave, transverse wall on the opposed, or other, side; said convex and concave walls being cambered in the direction of rotation of said first compression stage; a second compression stage; means for rotating said first stage at a peripheral velocity $+U_1$ and said second stage at a peripheral velocity $-U_2$ having a fixed ratio to $+U_1$; a plurality of compression channels in said second stage, each having an upstream portion defined by two side-walls and inwardly converging toward each other leading and lagging flat transverse walls inclined in the direction of rotation of said second stage, the degree of camber, or curvature, of said first convex wall and of said second concave wall of the first stage and the degree of inclination of the flat walls of the second stage being proportioned to discharge said fluid from said first stage and into said second stage at a relative entry velocity into said second stage parallel to the leading flat wall of the second stage and at an angle to the flat lagging wall of the second stage when said means is rotating said first and second stages at their respective normal operating peripheral velocities $+U_1$ and $-U_2$, said flat lagging wall of the second stage producing at least an oblique shock in said fluid upon the entry of said fluid into said second stage.

22. The compressor as defined in claim 21 in which the downstream portion of the compression channels of the second compression stage are shaped as diffusion channels.

23. The compressor as defined in claim 15 in which the downstream portions of the flow-accelerating channels of the stationary stage terminate in supersonic nozzles.

24. A centripetal flow compressor including at least first and second concentric rotors, each rotor including first and second side-discs, said side-discs each having an axial opening for accommodating bearing means for rotatively supporting said first and second side-discs on a frame of said compressor, and a hollow mechanical member axially separating and inter-connecting the outer peripheries of said first and second side-discs, said member including at least one centripetal flow compression stage rotatable with its rotor.

25. A centripetal compressor comprising at least first and second contra-rotatable rotors; each of said rotors having: a first side-disc rotatively supporting one end of said rotor, a second side-disc rotatively supporting the other end of said rotor, at least one centripetal flow compression stage having a plurality of blades oriented to produce centripetal compression of fluid when said compressor is in operation, said blades being supported by and being coupled to said first side-disc at one end of said blades, and a hollow rotatable member supporting with its adjacent end, and being coupled to, the other end of said blades, said hollow member extending along the rotational axis of said compressor and being coupled to the second side-disc of said rotor, and bearing means for rotatively supporting said first and second side-discs.

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