Sept. 23, 1958 M. W. BEARDSLEY 2,853,227

SUPERSONIC COMPRESSOR

Filed May 29, 1948 2 Sheets-Sheet 1

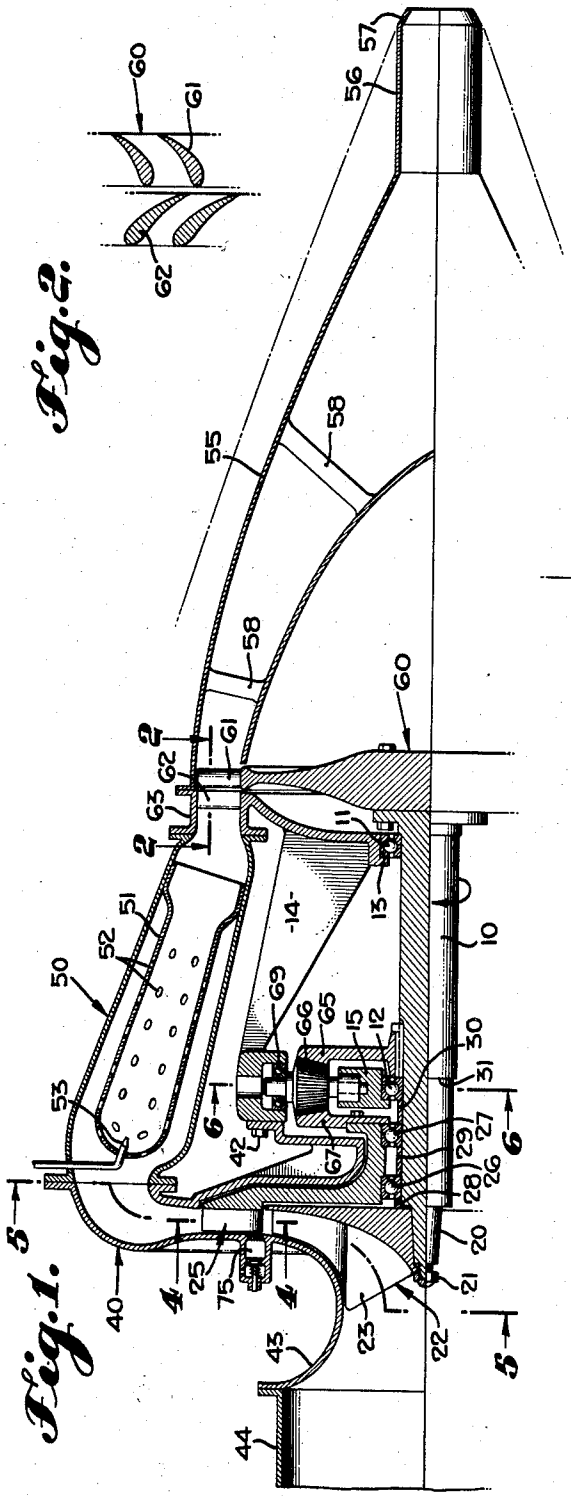


Fig. 1.

Fig. 2.

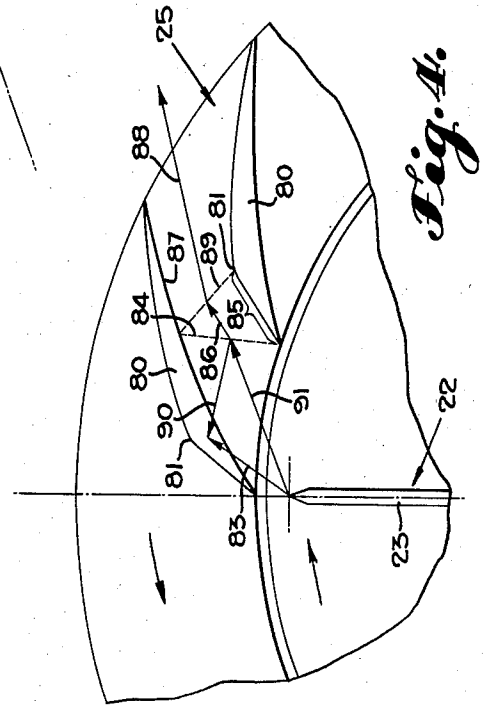


Fig. 4.

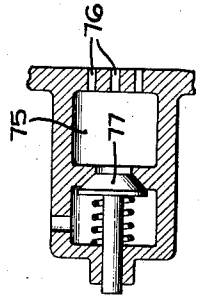


Fig. 3.

MELVILLE W. BEARDSLEY,
INVENTOR.

BY Robert W. Fulwider
ATTORNEY

Sept. 23, 1958 M. W. BEARDSLEY 2,853,227

SUPERSONIC COMPRESSOR

Filed May 29, 1948 2 Sheets-Sheet 2

Fig. 5.

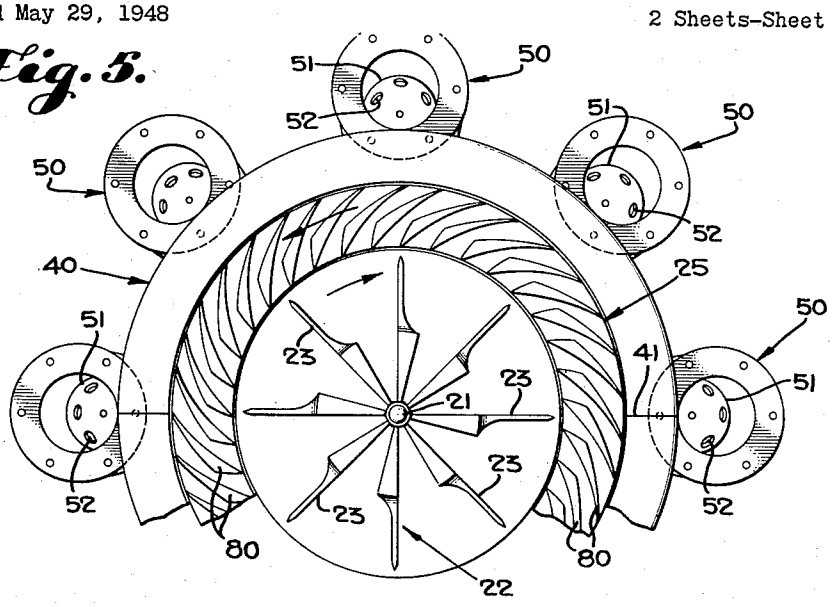
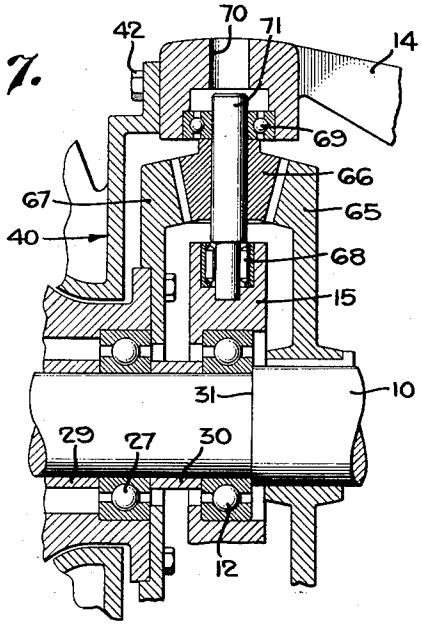


Fig. 7.



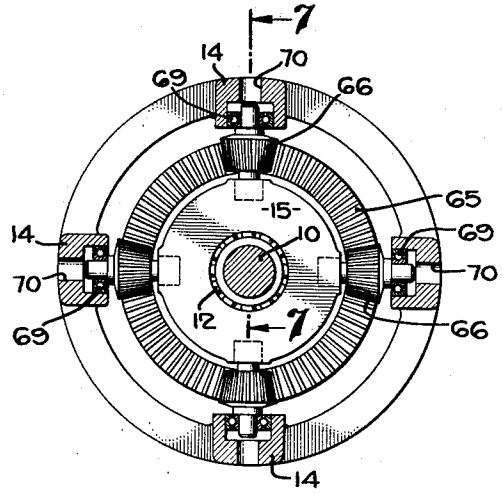


Fig. 6.

MELVILLE W. BEARDSLEY,
INVENTOR.

BY
Robert B. Fulwider
ATTORNEY

United States Patent Office 2,853,227 Patented Sept. 23, 1958

2,853,227

SUPERSONIC COMPRESSOR

Melville W. Beardsley, Venice, Calif.

Application May 29, 1948, Serial No. 29,971

8 Claims. (Cl. 230—114)

My invention relates generally to fluid compressors and more particularly to such compressors which are adapted to deliver compressed gas at a relatively high rate of flow with a relatively low ratio of compression as compared to expansible chamber type compressors. The novel compressor described herein is, however, capable of higher compression ratios than the conventional centrifugal compressor while maintaining the relatively high rate of flow of the centrifugal type.

The high flow type of compressor just described is used, for example, to compress air prior to its introduction into the combustion chamber of a jet propulsion engine. While the compressor embodying my invention is illustrated herein as used in connection with a turbo-jet engine, it will be realized by those skilled in the art that the usefulness of my invention is not confined to such engines, but has wide application in other fields, for example, in the compressing of air for industrial uses, in the firing of various types of furnaces, in the first stages of apparatus for producing liquified gases, and in many other fields.

Basically, turbo-jet reaction engines comprise a compressor, a combustion chamber into which the compressed air delivered by said compressor is introduced together with a liquid or gaseous fuel, a gas turbine actuated by the combustion products exhausting from said combustion chamber, and a jet or tail pipe for exhausting the combustion products after passing through said turbine and for producing the desired reactive jet. The turbine provides the power to drive the compressor.

Conventional practice in the design of turbo-jet engines has been to employ either centrifugal or axial flow compressors, directly driven, or driven through a gear train by the gas turbine. Briefly, the centrifugal compressors employed in engines of the type described employ a relatively rapidly rotating rotor and a set of stationary diffuser blades located surrounding the periphery of the rotor and adapted to convert the kinetic energy of the centrifugally accelerated air into compression thereof.

The axial flow compressor comprises essentially an alternating series of rotating and stationary blades, alternately inclined whereby to force the air in an axial direction through the compressor, the stationary blades serving as diffusers and to compress the air.

Both of the above described types of compressors have certain disadvantages, the centrifugal type having a relatively low efficiency, and the axial flow type being extremely intricate and difficult to design and construct. The entrance and exit velocities of gas flowing through both the radial (centrifugal) and axial flow compressors have usually been less than the velocity of sound, i. e., at velocities of a Mach number less than 1.0.

It is well known that a compressible fluid moving through a conduit at rates well below Mach 1.0 reacts in accordance with Bernoulli's theorem when encountering a constriction, i. e., the pressure of the fluid decreases as the cross-sectional area of the conduit decreases. Conversely, as the conduit cross-section increases, the pressure of the moving fluid increases. This change in pressure is due to the fact that, in order for the fluid to all pass through the conduit, the relative velocities of the fluid in various parts of the conduit must differ, the velocity being greatest at the throat or most constricted portion of the conduit.

At fluid velocities of greater than Mach 1.0, however, the Bernoulli behavior is essentially reversed. That is, the pressure of the gas increases and the velocity decreases as the cross-sectional area of the conduit becomes less.

This principle is made use of in the ram jet reactive engine of the type designed to move at velocities greater than Mach 1.0. Here the compressor comprises simply a forwardly facing constricted opening, and the result of the forward motion of the engine is to compress the air entering through the forwardly facing opening.

It will be seen first of all that if a centrifugal compressor capable of accelerating air to a velocity greater than Mach 1.0 were designed and a set of stationary diffusers placed around the periphery of the rotor and into which the air was centrifugally discharged from the rotor, the device would serve to reduce the velocity and compress the air thus discharged. The essential difficulty with the arrangement just described is, however, that in order to produce a rotor capable of accelerating air to supersonic velocities, the extreme centrifugal stresses encountered present nearly insurmountable difficulties.

It will be realized, however, that the behavior of gases moving through conduits is essentially due to the relative motion between the gas and the conduit walls. Thus, if a converging conduit is moved at a velocity over Mach 1.0 through stationary air, the result is to compress the air entering the conduit as in the ram jet engine above described. Compression is also the result if air moving at a velocity over Mach 1.0 is introduced into a stationary conduit. Ergo, the same result is achieved if the sum of the velocities of a conduit moving through oppositely moving air is greater than Mach 1.0. My novel compressor employs the last described principle.

Briefly described, I employ a pair of counter-rotating impellers, coaxially arranged, the innermost rotor acting in the manner of a conventional centrifugal blower to radially and tangentially accelerate air passing therethrough. From the inner impeller the air is discharged in a generally tangential direction into the blades of the outer rotor (hereinafter called diffuser rotor) which, as previously stated, is moving in a direction counter to the impeller rotor. The blades of the diffuser rotor are centrally thickened so that the passage between each pair of successive blades has an essentially converging-diverging or Venturi shaped constriction.

Bearing in mind the foregoing discussion, then, it is a major object of my invention to provide a fluid compressor having, relatively, a much greater efficiency than can be achieved in conventional centrifugal compressors.

Another object of my invention is to provide a compressor of the class described which is capable of a higher ratio of compression than can be achieved in conventional rotary compressors.

It is a further object of my invention to provide a rotary compressor in which the rotating members turn at a relatively low rate whereby to avoid extreme centrifugal stresses.

A still further object of my invention is to provide a compressor of the class described which is well adapted for use in connection with reaction jet engines for aircraft.

The foregoing and other objects and advantages of my invention will become apparent from a consideration of the following detailed description of one embodiment

thereof, such consideration being given also to the accompanying drawings, in which:

Figure 1 is a partially sectioned, elevational view taken through a turbo-jet engine embodying a compressor according to my invention;

Figure 2 is a horizontal section taken on the line 2—2 of Figure 1;

Figure 3 is an enlarged elevational section of a portion of the device illustrated in Figure 1;

Figure 4 is an enlarged elevational section take on the line 4—4 in Figure 1;

Figure 5 is a fragmentary elevational section taken on the line 5—5 in Figure 1;

Figure 6 is an elevational section taken on the line 6—6 in Figure 1; and

Figure 7 is an enlarged elevational section taken on the line 7—7 in Figure 6.

Throughout the drawings, the reference character **10** identifies a main rotary shaft turning in the direction indicated by the surrounding arrow in Figure 1. The shaft **10** is supported for rotation in a pair of radial ball bearings **11** and **12**. The rearward bearing **11** is secured in a suitable recess **13** in a frame **14**, and the forward bearing **12** is supported within non-rotating collar member **15**, the nature of, and the support for which will be hereinafter described.

Secured to the forward end of the shaft **10** by means of a taper stub **20** and a nut **21** is a multiple bladed impeller **22** having eight radial blades **23**.

A diffuser rotor **25** is rotatably mounted on the forward end portion of the shaft **10** and is coaxially positioned with respect to the impeller **22**. The rotatable mounting of the diffuser rotor **25** is provided by a pair of ball bearings **26** and **27**. A number of collars **28**, **29** and **30**, positioned on the shaft **10** between the impeller **22**, the bearings **26**, **27** and **12**, serve to separate and prevent axial displacement of the bearings, being held against a shoulder **31** in the shaft **10** by the forward attachment nut **21**.

Surrounding the impeller **22** and the diffuser rotor **25** is a stationary housing or manifold **40**, which is horizontally divided as indicated by the reference character **41** in Figure 5 whereby the housing **40** may be assembled over the impeller **22** and the diffuser rotor **25** after the latter have been assembled onto the shaft **10**. The housing **40** is secured to the frame **14** by means of a number of bolts **42**. An outward flared portion **43** in the housing **40** is secured by suitable flanges to the rearward end of an air-induction conduit **44**.

Eight combustion chambers **50** are mounted circumferentially around the axis of the shaft **10**, being supported at their forward ends by suitable flanges secured to the housing **40** and at their rearward ends by suitable flanges secured to the frame **14**. Each of the combustion chambers **50** includes an interior liner member **51** welded by a skirt-like projection at its rearward end to the wall of the chamber **50**. The liner member **51** in each of the combustion chambers **50** is provided with a number of apertures **52** through which air passes inwardly into the interior of the liner **51**.

Atomized liquid fuel is introduced at the forward end of the liner member **51** by means of a nozzle **53**, combustion taking place inside the liner. Since the principles of combustion and jet propulsion are not involved in my invention, a detailed description of the engine per se is not felt to be necessary herein.

A double walled frusto-conical conduit is secured by a suitable flange to the frame **14** and serves to conduct the products of combustion from the combustion chambers **50** rearwardly, exhausting the same through a suitable jet pipe **56** provided at its rearward end with an inturned lip portion **57**. The inner and outer walls of the conduit **55** are separated by suitably positioned and streamlined radial struts **58**.

Secured to the rearward end of the shaft **10** is a turbine wheel **60** having a number of streamlined radial blades **61** at its outer periphery which are interposed in the stream of combustion products exhausting from the combustion chamber **50**. A series of stator blades **62** are provided just ahead of the turbine blades **61**, the stator blades being counter-inclined, as indicated in Figure 2, serving the additional purpose of supporting the outer portion **63** of the frame **14**.

As has been previously described, it is desired that the diffuser rotor **25** rotate in the opposite direction from that of the impeller **22**. To this end, a reversing gear train is provided which comprises a large bevel gear **65** keyed to the shaft **10** immediately behind the shoulder **31**, a series of small bevel idler gears **66**, supported between the frame **14** and the collar member **15**, and a forward large bevel gear **67** bolted to the rearward surface of the diffuser rotor **25**. The bearings for the idler gears **66** include roller bearings **68** fixed in the collar member **15** and ball bearings **69** secured in the frame member **14**, as illustrated in Figure 7. A number of radial bores **70** are provided in the frame **14** through which the shaft **71** of each of the idler gears **66** may be assembled. It will be noted that the radial shafts **71** serve to support the collar member **15**.

Thus it will be seen that the direction of rotation of the impeller **22** is clockwise, while the direction of rotation of the diffuser rotor **25** is counter-clockwise, as indicated in Figure 5.

In order to initiate the flow of air through the impeller **22** when starting the compressor, it is, in some instances, necessary to relieve the pressure in the throat of the diffuser rotor **25** until the rate of air flow through the compressor approaches certain critical velocities, as will be hereinafter described. To this end, an annular manifold chamber **75** is formed in the housing **40** and communicated with the annular area adjacent the throat of the diffuser **25** by a number of passages **76**, as shown in Figure 3. A number of spring-loaded valves **77** are provided at intervals around the manifold **75** whereby to relieve the initial pressure in the manifold **75** until the flow of air through the compressor reaches or approaches critical rates as will be hereinafter described.

For a discussion of the behavior of the air passing through the compressor, reference should now be had to Figure 4. Here a portion of the outer periphery of the impeller **22** is shown, having thereon the outer end of one of the impeller blades **23**. A portion of the diffuser rotor **25** is also shown in Figure 4, together with an adjacent pair of diffuser blades **80**, the latter being supported on the body of the diffuser rotor **25** and projecting forwardly therefrom.

As can be seen best in Figure 4, each of the diffuser blades **80** is thickened in a portion generally midway between its ends whereby to produce a constriction point **81** on the leading surface of each blade **80**. Thus the passage between each pair of successive blades has a generally hour-glass shape, being narrowest at a point adjacent the constriction point **81**.

Air discharged from the impeller **22** enters the space between the diffuser blades **80** in a direction indicated by the line **83** in Figure 4. The line **83** represents the direction of the resultant vector derived from the relative radial and tangential velocities of the air exiting from the impeller **22**. The velocity of the air in the direction indicated by the arrow **83** is somewhat less than Mach 1.0.

The speed of the diffuser rotor **25** is such that the sum of the velocity of the air indicated by the arrow **83** and the tangential velocity of the diffuser blades **80**, as shown by the arrow **90**, is in the resultant direction of the arrow **91**, and is somewhat greater than Mach 1.0. Thus the air, upon encountering the blades **80** behaves in accordance with the principles of supersonic flow. One of such principles is that when the direction of air flowing at supersonic velocities is changed, this change is not gradual but occurs suddenly along a front or surface known as

a "shock plane" or "shock surface." These shock planes may be photographed due to the radical difference in density of the air on the two sides of the plane. It is also observed that the density on the downstream side of the shock plane is greater than that on the upstream side, and the flow velocity consequently lower.

The effect of the supersonic velocity encounter between the diffuser blade **80** and the discharging air from the impeller **22** is to first produce a shock plane identified in Figure 4 by the dotted line **84**, the shock plane **84** being produced by the impact of the discharging air against the forward blade surface portion **85** between the leading edge of the blade **80** and the constriction point **81**. After passing through the shock plane **84**, the air is traveling in a direction indicated by the arrow **86**, and upon encountering the adjacent blade at its rearward surface **87**, again changes its direction as indicated by the arrow **88**. Since the air is still traveling at supersonic velocity (relative to the blades **80**), this second change of direction produces a second shock plane **89** which crosses the area between two adjacent blades at approximately the constriction point **81**.

It will be recalled that the velocity of the air is reduced each time that it passes through a shock plane and the net result of passing through the two shock planes **84** and **89** is to reduce the relative velocity of the air in the diffuser passage to a value near Mach 1.0. Therefore, the air thereafter behaves in accordance with Bernoulli's theorem and the widening space between the adjacent diffuser blades **80** results in further compression of the air. Still further compression is achieved by the continual widening of the manifold passage within the housing **40** as the air is carried to and discharged into the combustion chambers **50**.

It has sometimes been found necessary, in order to initiate the flow of air through the compressor, to provide the manifold **75** connected by passages **76** to the throat area of the diffuser **25**. After the initial flow has been instituted, the pressure in the manifold **75** drops, the valve **77** closes, and the operation thereafter proceeds in accordance with the supersonic principles just described.

It will be realized by those skilled in the art that the counter-rotation of the impeller **22** and the diffuser **25** may be accomplished in various ways. For example, in stationary installations, a pair of electric motors may be employed to drive these two rotary members in opposite directions. Another method of achieving counter-rotation is by the provision of two instead of one turbine wheel, such as that indicated by the reference character **60** in Figure 1. Two such turbine wheels can be provided with oppositely inclined blades whereby to rotate in opposite directions. One of the turbine wheels is then mounted on a coaxial hollow shaft and connected to drive the diffuser rotor.

It is believed obvious, furthermore, that the principles of supersonic compression involved in the foregoing construction are equally applicable to axial flow compressors. In such devices, a series of counter-rotating impellers and diffusers can be arranged along an axis to accomplish the same supersonic relative velocity as is produced in the diffuser of the centrifugal type of compressor illustrated herein.

While the compressor shown and described herein is fully capable of achieving the objects and providing the advantages hereinbefore stated, it will be realized that it is capable of considerable modification by those skilled in the art. Therefore, I do not mean to be limited to the form shown and described herein, but rather to the scope of the appended claims.

I claim:

1. In a fluid compressor of the class described, the combination of: a rotatably mounted drive shaft; a bladed impeller secured to a forward end of said shaft for rotation therewith; a diffuser comprising a body rotatably mounted on said shaft adjacent said impeller and rearwardly thereof, and a plurality of blades on said body projecting forwardly therefrom around the periphery of said impeller, said diffuser blades having sharp leading edges and being inclined to the radius of said impeller and thickened at points therein opposite a point in the next adjacent blade intermediate the leading and trailing edges thereof whereby to form a plurality of Venturi-like passages with converging entrance sections between said blades; and drive means to rotate said shaft and diffuser in opposite directions whereby air centrifugally discharged from said impeller enters said Venturi passages at a relative velocity above the speed of sound in air at the entrances of said passages.

2. A compressible fluid compressor comprising: a centrifugal fluid impeller mounted for rotation on an axis to discharge fluid tangentially outward from said axis; a diffuser having a coaxial rotor mounted adjacent said impeller and a plurality of blades on said rotor projecting therefrom around the periphery of said impeller in the path of fluid discharging as aforesaid, said blades each having sharp leading edges and being inclined to a radius of said impeller and having a thickened portion opposite a point in the next adjacent blade substantially to the rear of the leading edge thereof, whereby to form a passage between each adjacent pair of said blades with a constricted throat in said passage and a substantial converging section preceding said throat; and drive means to rotate said diffuser and impeller at such relative speed that compressible fluid discharged as aforesaid from said impeller enters said passages at a velocity relative thereto which is above the speed of sound in the fluid in the region between said impeller and passages.

3. The construction of claim 2 further characterized by having a stationary wall adjacent said diffuser positioned to laterally enclose said passages, and starting pressure relief means including at least one controllable port in said wall to selectively release fluid from said region whereby to initiate flow of said fluid at said speed.

4. A compressible fluid compressor comprising: a centrifugal fluid impeller mounted for rotation on an axis to discharge fluid tangentially outward from said axis; a diffuser having a coaxial rotor mounted adjacent said impeller and a plurality of blades on said rotor projecting therefrom around the periphery of said impeller in the path of fluid discharging as aforesaid, said blades each having a bulge formed on the advancing surface thereof, said bulge rising to a maximum thickness at a point opposite a point in the blade adjacent said surface, substantially to the rear of the leading edge thereof, whereby to form a passage between each adjacent pair of said blades with a constricted throat in said passage and a substantial converging section preceding said throat; and drive means to rotate said diffuser and impeller at such relative speed that compressible fluid discharged as aforesaid from said impeller enters said passages at a velocity relative thereto which is above the speed of sound in the fluid in the region between said impeller and passages.

5. A compressible fluid compressor comprising: a fluid impeller mounted for rotation on an axis to discharge fluid tangentially outward from said axis; a diffuser having a coaxial rotor mounted adjacent said impeller and a plurality of blades on said rotor projecting therefrom around the periphery of said impeller in the path of fluid discharging as aforesaid, said blades each having sharp leading and trailing edges and each being inclined to a radius of said impeller and each having a thickened portion intermediate the leading and trailing edges thereof, whereby to form a passage between each adjacent pair of said blades with a constricted throat in said passage, said throat being positioned to provide a substantial converging section preceding said throat, and a substantial diverging section following said throat; and drive means to rotate said diffuser and impeller at such relative speed that compressible fluid discharged as afore-

said from said impeller enters said passages at a velocity relative thereto which is above the speed of sound in the fluid in the region between said impeller and passages.

6. A compressible fluid compressor comprising: a centrifugal impeller; a diffuser having a coaxially mounted rotor adjacent said impeller and sharp edged blades on said rotor arranged in a circular series surrounding said impeller to provide a plurality of discharge passages between adjacent pairs of said diffuser blades, said blades being thickened intermediate their edges to provide converging entrance portions in said passages; means to rotate said impeller at a predetermined speed; and means to counter-rotate said rotor at a speed such that the relative periphery velocity of said impeller and diffuser blades at the point of closest proximity thereof is greater than the speed of sound in said fluid in the region of said point.

7. The construction of claim 6 further characterized by having lateral walls enclosing said passages and an initial pressure relief port in at least one of said walls.

8. The construction of claim 7 further characterized by having a pressure-responsive valve in said port to close the same when said initial pressure drops below a predetermined value.

References Cited in the file of this patent

UNITED STATES PATENTS

956,839	Wiedemann	May 3, 1910
1,091,581	Ljungstrom	Mar. 31, 1914
1,462,592	Bentley	July 24, 1923
2,075,984	Handforth	Apr. 6, 1937
2,317,723	Becht	Apr. 27, 1943
2,318,990	Doran	May 11, 1943
2,344,366	Price	Mar. 14, 1944
2,383,385	Heintze	Aug. 21, 1945
2,400,714	Rowledge	May 21, 1946
2,404,334	Whittle	July 16, 1946
2,435,236	Redding	Feb. 3, 1948
2,575,682	Price	Nov. 20, 1951
2,664,850	Smith	Jan. 5, 1954

FOREIGN PATENTS

537,720	Great Britain	July 3, 1941