Dec. 15, 1970 P. G. SCHWAAR 3,546,880

COMPRESSORS FOR GAS TURBINE ENGINES

Filed Aug. 4, 1969 3 Sheets-Sheet 1

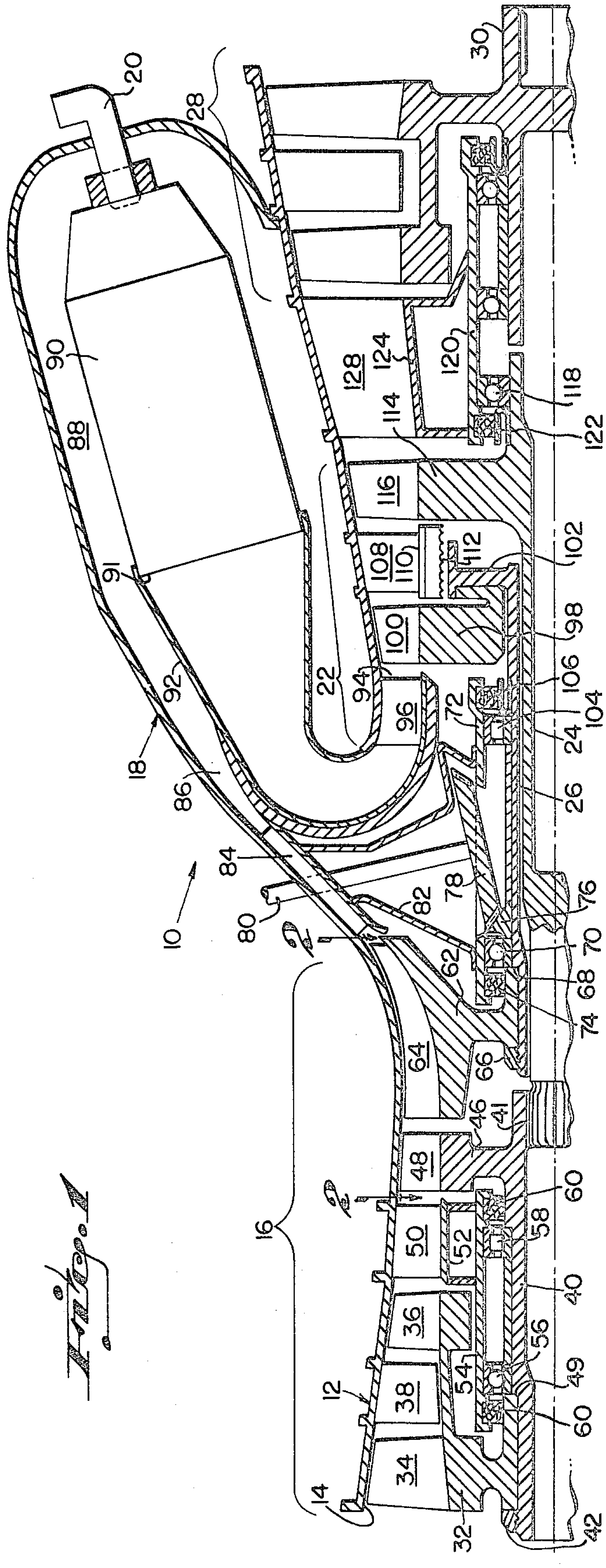


Fig. 1

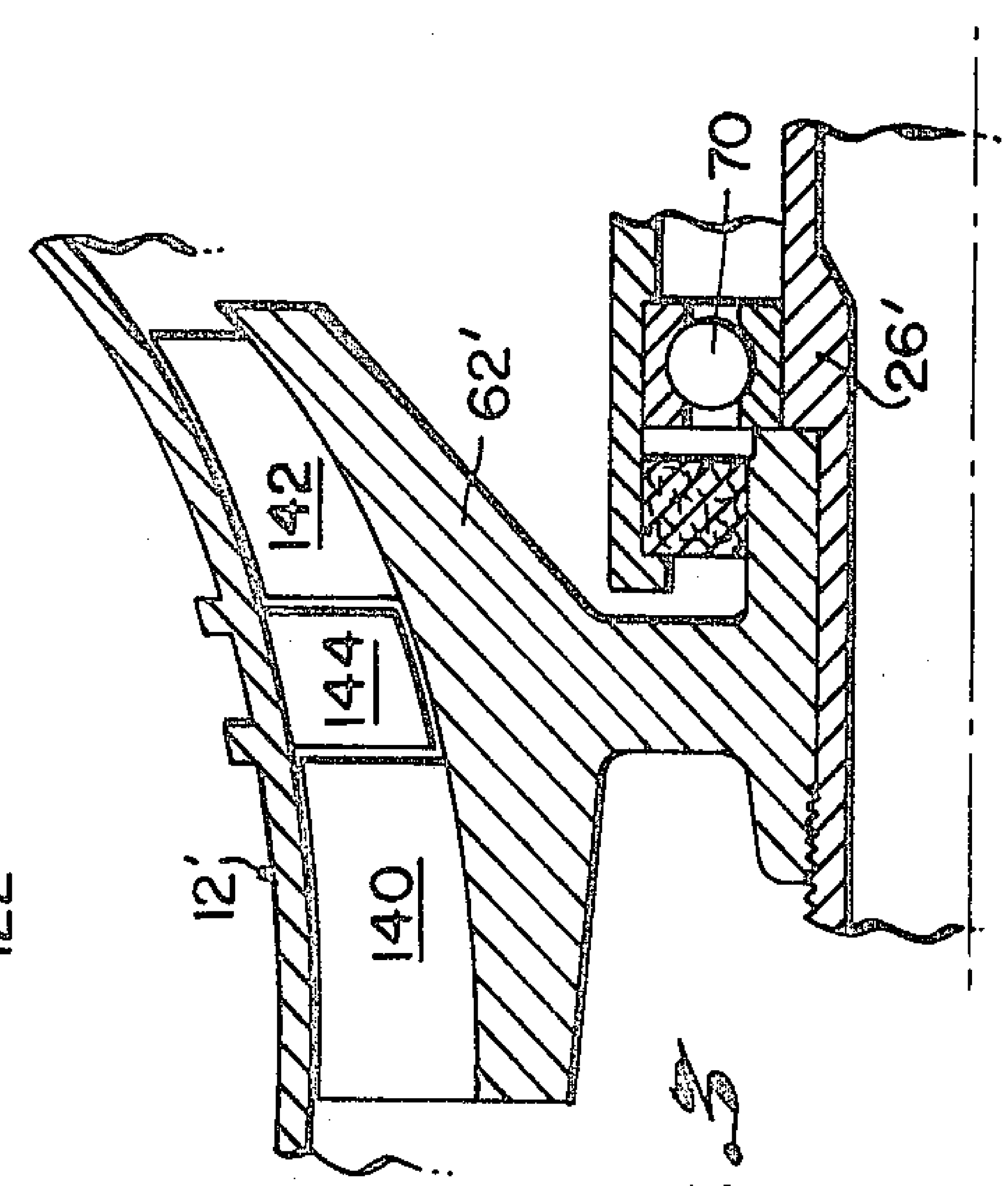


Fig. 3

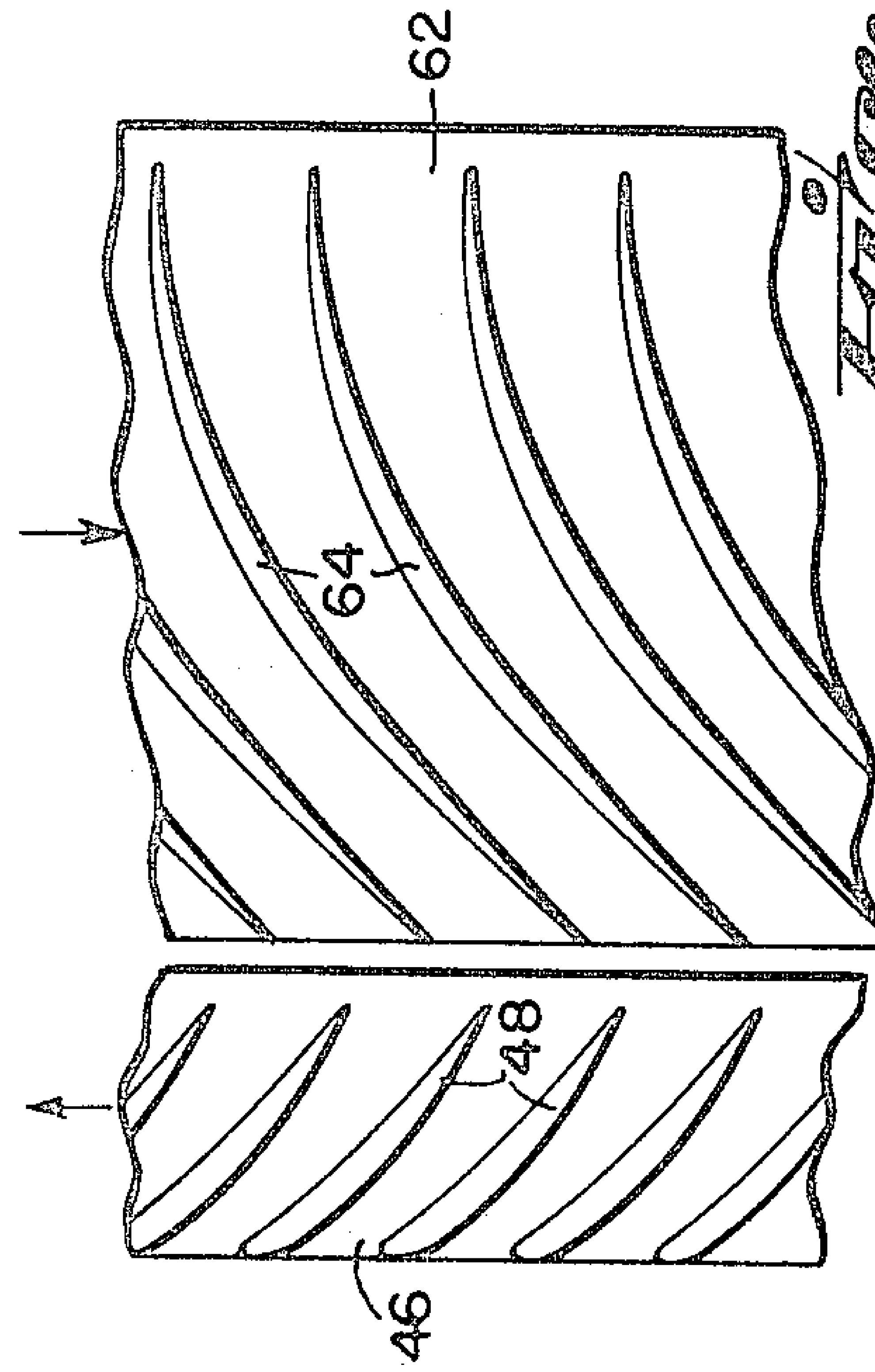


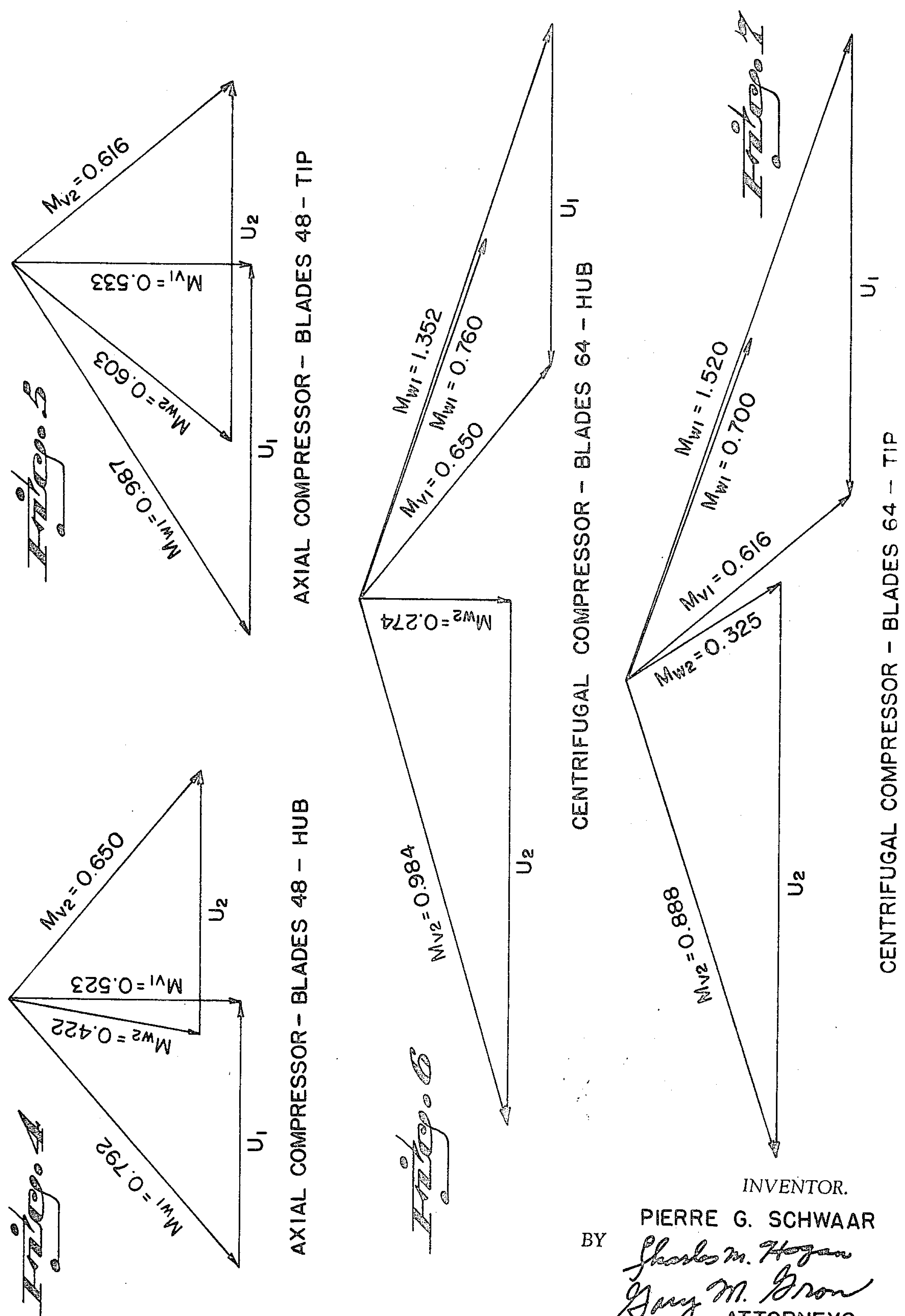
Fig. 2

INVENTOR.
PIERRE G. SCHWAAR

BY Charles M. Hogan
Gary M. Gron
ATTORNEYS.

COMPRESSORS FOR GAS TURBINE ENGINES

Filed Aug. 4, 1969

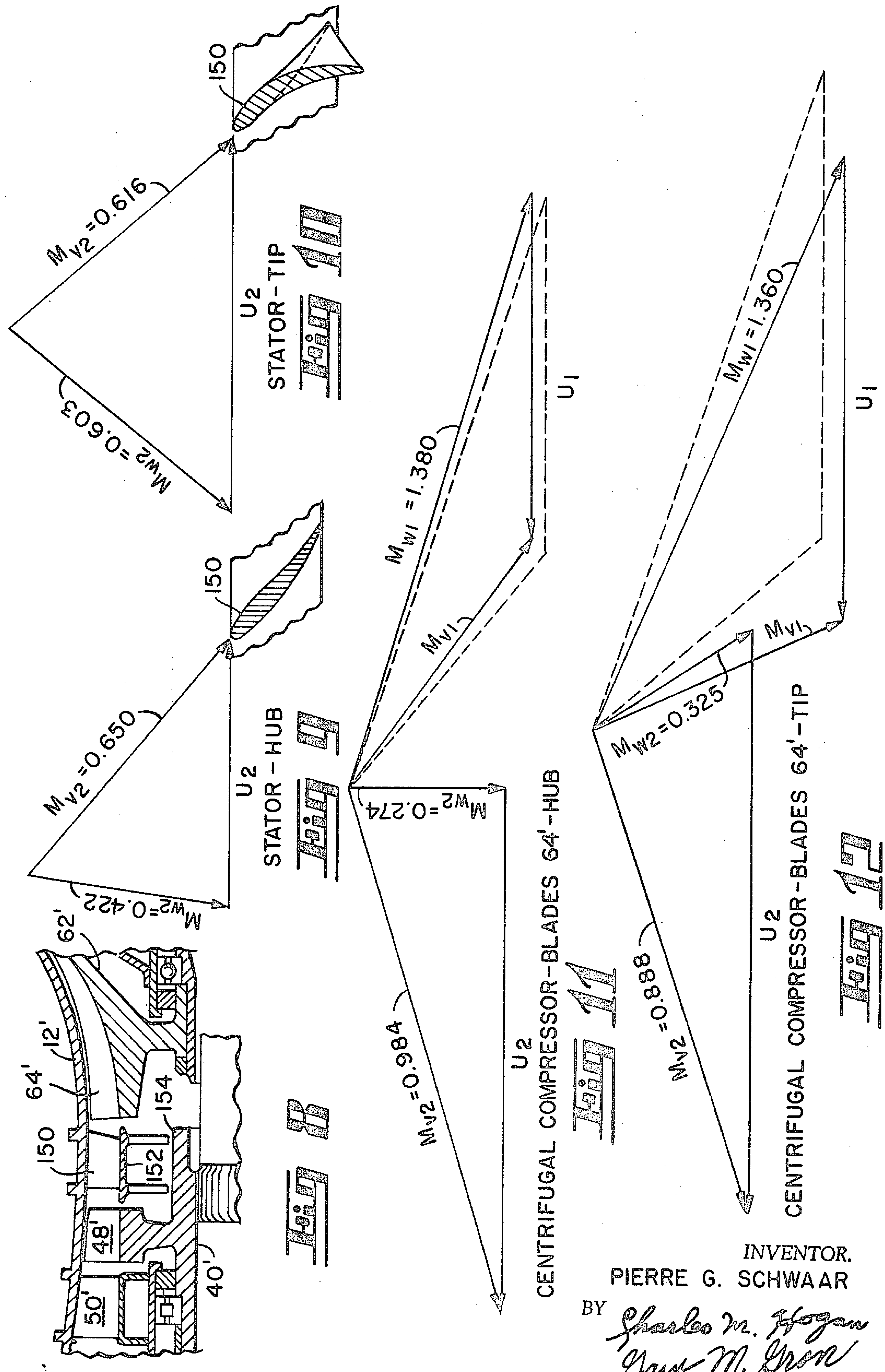


INVENTOR.
PIERRE G. SCHWAAR
BY
ATTORNEYS.

Dec. 15, 1970 P. G. SCHWAAR 3,546,880

COMPRESSORS FOR GAS TURBINE ENGINES

Filed Aug. 4, 1969 3 Sheets-Sheet 3



INVENTOR.
PIERRE G. SCHWAAR
BY Charles M. Hogan
Gary M. Gron
ATTORNEYS.

3,546,880

COMPRESSORS FOR GAS TURBINE ENGINES

Pierre G. Schwaar, Monroe, Conn., assignor to Avco Corporation, Stratford, Conn., a corporation of Delaware

Continuation-in-part of application Ser. No. 700,956, Jan. 26, 1968. This application Aug. 4, 1969, Ser. No. 857,264
Int. Cl. F02c 7/04
U.S. Cl. 60—39.16 14 Claims

ABSTRACT OF THE DISCLOSURE

The disclosure illustrates a gas turbine engine having an improved compressor comprising a three-stage axial flow compressor and a counterrotating supersonic mixed flow compressor immediately adjacent the downstream end of the axial flow compressor. The axial flow compressor and the mixed flow impeller are rotated at speeds that cause the air discharged from the axial flow compressor to enter the mixed flow impeller at a relative velocity greater than sonic. Shock waves are then set up in the inlet portion of the impeller which reduces the relative velocity to a subsonic value and increases the pressurization of the air. In the portion of the impeller adjacent the discharge end the air is further pressurized by diffusion and centrifugation. An alternate embodiment utilizes a stator vane assembly between the axial flow compressor and the mixed flow impeller. The stator vane assembly is shaped to minimize the velocity variation from the hub to the tip of the mixed flow impeller entrance.

This is a continuation-in-part of application Ser. No. 700,956 filed Jan. 26, 1968, now abandoned.

The present invention relates to compressors and more specifically to compressors for use in gas turbine engines.

It is well known that one of the parameters reflecting the performance of a gas turbine is the pressure ratio of the compressor, or in other words the relative increase in pressure imparted to the air supplied to the combustor. An increase in pressure ratio results in an improvement of the specific fuel consumption if the aerodynamic efficiency of the compressor can be maintained at the same level.

One way to achieve a high pressure ratio is to utilize an axial flow compressor, commonly used in large gas turbine engines. It is well known in the art that this compressor has a high degree of efficiency and is capable of producing a very high pressure ratio if enough compression stages are provided. However, the large number of stages increases the complexity and the cost of the axial compressor to a level that practically negates its use in a low cost, small gas turbine.

To achieve high pressure ratios with a high degree of simplicity and resultant decrease in cost, the centrifugal compressor has been extensively used. In this type of compressor high pressure ratios are achieved by rotating the impeller at such a speed that the air is discharged from the impeller at a supersonic velocity level. It is then necessary to provide a diffuser which reduces the velocity of the air to a subsonic value by a shock wave process and further reduces the velocity to the lower level used in the combustion portion of the engine. While this type of compressor has been used quite extensively for small gas turbine engines, it suffers from a relatively low efficiency, owing to the fact that the shock process associated with supersonic flow in the diffuser is difficult to properly concontrol, and the area over which the flow is supersonic and the shock process takes place, is relatively large because of the large diffuser entrance diameter. This greatly

increases the diffuser losses and substantially reduces the efficiency of such a compressor.

Accordingly, it is an object of the present invention to provide simple means to design an efficient and compact compressor unit capable of very high pressure ratios, especially suitable for small gas turbine applications.

The above ends are achieved by an axial flow compressor rotating in a first direction for discharging air at an absolute velocity lower than the speed of sound. A counterrotating bladed impeller having a generally axially directed inlet flow path receives air from the axial flow compressor. The rotational speed of the bladed impeller is sufficient to produce a relative velocity between the discharge air and the impeller inlet greater than the speed of sound, thereby producing shock waves adjacent the inlet of the impeller to reduce the absolute velocity of air below sonic and pressurize it. The impeller has an outlet flow path spaced radially outward from the inlet whereby air is substantially radially accelerated and diffused for further increase of its pressure. As a result, the flow area over which the air flow is supersonic is minimized.

In one aspect of the invention a compressor of the above general configuration has a series of stator vanes positioned between the axial flow compressor and the bladed impeller. The stator vanes are shaped to turn the flow discharged at the bladed impeller in such a way that the radial variation in relative velocity from the impeller to the tip of the blades on the impeller is minimized.

In the drawings:

FIG. 1 is a longitudinal cross section of a gas turbine engine having a compressor unit embodying the present invention;

FIG. 2 is a view taken on lines **2—2** of FIG. 1 and illustrating a feature of the compressor of the engine of FIG. 1;

FIG. 3 is a fragmentary longitudinal view of a portion of a compressor, such as that shown in FIG. 1, illustrating an alternate embodiment of the present invention;

FIGS. 3, 4, 5, 6 and 7 are velocity triangles showing the Mach number and direction of the flow through the compressor of FIG. 1;

FIG. 8 is a fragmentary view of an alternate embodiment of a compressor constructed in accordance with the present invention; and

FIGS. 9, 10, 11 and 12 are velocity triangles showing the Mach number and direction of the flow through the compressor of FIG. 8.

Reference is now had to FIG. 1 which shows a gas turbine engine **10** comprising an annular housing **12** having an inlet **14** for passage of air to a compressor unit **16**. As later described in detail, the compressor unit **16** has rotating portions which pressurize air and discharge it through an annular flow path to a combustion unit, generally indicated by reference numeral **18**. The pressurized air is then mixed with fuel injected into the combustion unit **18** by a fuel nozzle **20** and the resultant mixture is ignited by well-known means to generate a hot gas stream which is discharged across a turbine unit **22**, wherein a portion of the energy of the hot gas stream is extracted to drive the counterrotating portions of the compressor unit **16** through interconnecting shafts **24**, **26**. From the turbine unit **22** the hot gas stream may be discharged through an exhaust nozzle to produce a propulsive thrust. As herein illustrated though, a power turbine unit **28** extracts a major portion of the energy available from the hot gas stream to drive a rotatable output shaft **30** which is used as the prime power output for the engine **10**.

In accordance with the present invention, the engine **10** incorporates an improved compressor unit **16**, as described in detail below, together with the cooperating engine components.

The compressor unit **16** has an upstream axial flow portion which comprises a rotor **32** having first and second stages of circumferentially positioned compressor blades **34** and **36**, respectively secured thereto by suitable retaining means (not shown). The rotor **32** is telescoped onto a shaft **40** and held against a shoulder **49** by a retaining nut **42**. The aft portion of the shaft **40** is integral with a rotor **46**, having a third stage of circumferentially positioned compressor blades **48**. The aft portion of the shaft **40** also has a splined opening **41** which is telescoped over a splined portion of the interconnecting shaft **24**.

A first series of circumferentially positioned compressor stator vanes **38** extend from the housing **12** into an annular space between the adjacent stages of blades **34** and **36**. A second series of circumferentially positioned compressor vanes **50** extend from the housing **12** radially inward between the second and third stages of compressor blades **36** and **48**, respectively. The compressor blades **50** are fabricated to provide a relatively rigid supporting structure for an inner annular structural member **52**.

The structural member **52** is secured to an annular bearing sump chamber **54**. A pair of bearings **56** and **58**, used to journal the shaft **40** are mounted in the sump chamber **54**. The ends of the sump chamber **54** are sealed by suitable annular seal assemblies **60** which may be of the friction seal or laybrinth seal type.

The downstream portion of the compressor unit **16** is comprised of a rotatable impeller comprised of a hub **62** which forms the inner bounds of a generally annular flow path thereacross. The outer bounds of the flow path is then defined by the adjacent portion of the engine housing **12**. A series of generally radially extending blades **64** are secured to the outer periphery of the hub **62** by suitable retaining means (not shown). The hub **62** is telescoped over the interconnecting shaft **26** and urged against a shoulder **68** by a retaining nut **66**. The end of the shaft **26** adjacent the hub **62** is journaled by means of a bearing assembly **70** which is mounted in a generally annular sump chamber **72**.

The sump chamber **72** is sealed at its forward end by a seal assembly **74**. The shaft **26** may be used for accessory drive purposes by securing a bevel pinion gear **76** over the shaft **26** to engage a bevel gear **78**. The gear **78** is secured to a shaft **80** which in most instances would be journaled in a generally annular support structure **82** extending from the sump chamber **72**. The annular support structure **82** is secured at its outer periphery to the inward edges of a plurality of circumferentially positioned diffuser vanes **84** which receive the air discharged from the impeller blades **64**.

A generally annular entrance duct **86** is secured to the aft ends of the diffuser vanes **84** and provides a flow path for air into the combustion unit **18**. The combustion unit **18** comprises an annular chamber **88** which may be formed integrally with the engine housing **12**, as herein illustrated. A combustor **90** is positioned in the chamber **88**. The combustor **90** may take the form herein illustrated of a reverse-flow cannular combustor assembly. The cannular burners briefly comprise a series of generally axially extending perforated cans positioned around the chamber **88**. They are interconnected circumferentially by a tube and discharge the hot gases into a piece **92** which turns the flow to an aft direction and effects the transition from the circular can outlets **91** into the full turbine entrance annulus in front of nozzle **94**.

The turbine nozzle **94** generally comprises a series of radially extending vanes **96** secured to the engine housing **12** and to the inner portion of the transition piece **92**. A turbine rotor **98** is positioned adjacent the turbine nozzle **94** and has a plurality of radially extending turbine blades **100** which form a first stage of the turbine unit **22**. The turbine rotor **98** is secured by suitable means to the shaft **26** at a flange **102**, integral therewith. The aft portion of the shaft **26** is journaled by a bearing assembly **104** secured in the annular sump chamber **72**. A suitable seal assembly **106** seals the aft portion of the sump chamber **72**.

A plurality of circumferentially positioned turbine vanes **108** are secured to the engine housing **12** downstream of the turbine blades **100**. The inner ends of the vanes **108** connect with an annular duct member **110** having a suitable gas seal **112** between the duct member **110** and the flange **102** of the shaft **26**.

A second turbine rotor **114** is positioned downstream of the turbine vanes **108** and has secured thereto a plurality of turbine blades **116**. The rotor **114** is integral with the shaft **24** and is journaled by means of a bearing assembly **118**, supported by a bearing sump chamber **120**. The sump chamber **120** is sealed at its forward end by a seal **122** and secured to a generally annular support member **124**. The support member **124** is connected at its outer edge to a series of structural vanes **128** extending to the engine housing **12**.

As shown in FIG. 1, the counterrotating axial and centrifugal compressor sections are directly driven by two mechanically independent turbine rotors **100** and **116**, which thus rotate in opposite directions. It is clear, however, that counterrotation of the two compressor sections also can be achieved by a gear driven by a single turbine rotor.

In operation, the axial compressor has a flow velocity at exit of the last axial rotor stage **48** that is subsonic relative to the fixed engine housing **12** (absolute velocity). The counterrotating impeller **62**, however, causes the flow velocity of air entering the impeller to be supersonic relative to the impeller (relative velocity). Shock waves are generated in the entrance region of the impeller blading, which decelerate the flow to a subsonic velocity level relative to the impeller. The blading **64** in this entrance region of the impeller is characterized by an essentially constant, or slightly increasing, mean radius, and by a moderate camber angle, so that flow separation induced by shock-boundary layer interaction is minimized. In the rear portion of the impeller blanding, the flow is gradually turned radially outwards and toward the relative axial direction, thereby experiencing a further pressure increase by centrifugation and additional relative deceleration.

The type of impeller illustrated herein is generally termed a mixed flow impeller. However, the shape of the impeller blades differs from the usual mixed flow configuration in that there is relatively little or no turning of the flow in the first portion of the blades, i.e., the portion where the supersonic flow is decelerated to a subsonic velocity level by the shock wave process, as shown in FIG. 2. In addition, the essentially axial flow channel in the first portion of the impeller is designed so that there is relatively little or no increase of the relative flow passage area through the first portion of the blading, in order to stabilize the shock configuration and to ensure maximum efficiency of the compression by the shock process.

As a result of the swirl imparted to the fluid by the last axial stage **48**, a substantial part of the impeller work is done in the impeller entrance region. This enables the impeller to be designed with a smaller exit diameter and a smaller absolute exit Mach number than would be otherwise required for an equivalent pressure ratio. Accordingly, the impeller is preferably designed with a mixed axial-radial flow path, that is with an annulus channel forming an angle with the axial direction substantially smaller than 90° at the impeller discharge diameter.

FIGS. 4, 5, 6 and 7 illustrate exemplary flow conditions that exist in a typical compressor embodying the present invention. It should be noted, however, that the description of these flow conditions is not intended to limit the scope of the present invention but to merely enable a clearer understanding of the concepts involved in its operation. The velocities illustrated are for a compressor which has a transonic axial flow portion having three stages. While it is not necessary to utilize a transonic axial flow compressor in the invention, the use of a transonic compressor enables a reduction in the number of stages necessary to achieve a given pressure ratio.

The nomenclature of the velocity vectors generally expressed in Mach numbers (M) are as follows:

M_{V1}=absolute entrance velocity
M_{V2}=absolute exit velocity
M_{W1}=relative entrance velocity
M_{W2}=relative exit velocity
U_1=entrance tangential velocity of rotating portion of compressor
U_2=exit tangential velocity of rotating portion of compressor
$\hat{M}_{W1}$=subsonic velocities adjacent the entrance of the mixed flow impeller after shock process.

As shown in FIGS. 4 and 5, the relative velocity at the tip of the third stage of blades **48** is just below sonic level (M_{W1}=0.984). The absolute velocity at exit of the stage of blades **48** is subsonic (M_{V2}=0.650 at hub and M_{V2}=0.616 at tip), but the relative velocity at entrance of the impeller is supersonic over the entire channel height (M_{W1}=1.352 at the hub, M_{W1}=1.520 at the tip section). The impeller decelerates the supersonic flow to a subsonic velocity level in its front portion by a shock process which, depending upon the entrance Mach number M_{W1}, may take the form of a single-shock, a multishock, or a so-called pseudo-shock configuration. This is indicated by the subsonic Mach numbers $\hat{M}_{W1}$=0.760 at the hub, and M_{W1}=0.700 at the tip section, which characterize the flow conditions downstream of the front portion of the impeller. The impeller finally discharges the flow into the diffuser **84** with a subsonic absolute velocity (M_{V2}=0.984 at the hub, M_{V2}=0.888 at the tip section). The smaller flow velocity at entrance of the diffuser, as compared to that which would be realized with a conventional centrifugal compressor, and the smaller diffuser entrance diameter, result in a substantial decrease of the friction losses through that blading element.

The above axial compressor and mixed flow impeller provide an extremely high pressure ratio for a relatively simple configuration. It is to be noted that the supersonic flow that is necessary to achieve high pressure ratios in a radial type compressor has been shifted from the diffuser entrance region to the impeller entrance region. This greatly reduces the area over which the flow is supersonic and, accordingly, reduces the friction losses associated with supersonic flow. It is also to be noted that because a substantial portion of the compression work is accomplished by the shock process in the inlet portion of the impeller, the radial distance of the discharge relative to the inlet can be minimized for a given impeller pressure ratio. Furthermore, the lower flow velocity at the discharge of the impeller enables the use of a more efficient diffuser.

When a compressor design in accordance with the present invention is operated to produce comparatively high relative Mach numbers at the entrance of the bladed impeller it is desirable to minimize the radial Mach number variation between the hub and the tip of the blades in order to obtain a more uniform static pressure ratio through the impeller shock system over the radial extent of the annular flow path into the impeller. This insures that radial equilibrium of the flow will be preserved through the shock system and the flow conditions in the downstream portion of the impeller will be accordingly improved.

This aerodynamic improvement is achieved by providing a row of static blades between the downstream end of the axial flow compressor and the inlet to the impeller, as shown in FIG. 8. FIG. 8 illustrates the last stage **48'** of an axial flow compressor. A plurality of stator vanes **150** are mounted at their outer edge to the engine casing **12'** and at their inner edges to an annular channel member **152**, defining the inner bounds of the annular flow path across the stator vanes **150**. In accordance with practices known in the art, the channel shaped member **152** is adapter to provide a seal in cooperation with an extension **154** of the rotor **40'** for the axial flow compressor.

As shown particularly in FIG. 9 the vanes **150** are shaped at their radially inner end to impart no turning of the air discharged from the axial flow compressor towards an axial direction. As shown in FIG. 9, the leading edge of vane **150** is approximately in line with the Mach number vector M_{V2} of the air discharged from the last stage of the axial flow compressor. The trailing edge of vane **150** turns the flow slightly towards a tangential direction. This enables the relative angle of the flow at the entrance to bladed impeller **62'** to be uniform over the radial extent of the annular flow path.

The vanes **150** are cambered, however, so that their tip section imparts a substantial turning towards the axial direction, as shown in FIG. 10. It is illustrated in this figure that the leading edge of the outer end of vanes **150** is approximately in line with the Mach number vector M_{V2} of air discharged from the last stage of the axial flow compressor. The blade is turned, however, so that the downstream end of the blade points towards an axial direction.

The camber of vanes **150** produces Mach numbers into the entrance of the impeller **62'** as shown in FIG. 11. It is apparent from these figures that the tangential Mach number component of air entering the centrifugal compressor at the tip of blades **64** has been reduced by the camber of the outer end of vanes **150**. This minimizes the increased relative Mach number of the air caused by the increased tangential velocity component of the blades **64'** at their inlet tip. This results in relative entrance Mach numbers substantially equal. In the stator vane illustrated the radially inner section of the vanes **150** is shaped to turn the flow towards the tangential direction in a sense opposite to that of the turning at the outer portion of the vanes **150**. This enables the relative angle of the flow at the entrance of the impeller to be essentially uniform and enables the shape of the front portion of the impeller blades **64'** to be essentially constant from hub to tip. It should be apparent that the radially inner portion of vanes **150** can be shaped to impart no turning to the air flow and still achieve constant relative Mach numbers at the inlet to impeller **62'**.

In the particular engine embodiment shown in FIG. 1 the combustion unit **18** is of the reverse flow type, which enables a substantial reduction in axial length of the interconnecting shafts **24** and **26**. By utilizing this type of construction the natural frequency of the interconnecting shafts is sufficiently high to be above the normal operating conditions of the engine and would eliminate any necessity to compensate for this particular phenomena. The reverse flow combustion unit **18** additionally enables a substantial reduction in axial length for the engine **10** which greatly facilitates its use in applications where axial length is at a premium.

While the impeller arrangement described above and illustrated in FIGS. 1 and 2 has a substantial pressure ratio, the arrangement of FIG. 3 enables an even greater pressure ratio to be achieved. In FIG. 3 there is illustrated a modified impeller **62'** which is secured to a shaft **26'**, journaled by a bearing assembly **70**. In this impeller there is shown a first series of radially extending blades **140** forming an inlet portion of the impeller and a second series of generally radially extending blades **142** forming an outlet portion of the impeller **62'**. A series of circumferentially positioned vanes **144** are secured to an engine housing **12'** and extend between the adjacent series of blades **140** and **142**. In this arrangement the blades **140** receive the relative supersonic flow from the axial flow compressor and causes the air flow to be shocked down, thereby increasing its pressure. The vanes **144** receive the flow and turn it towards an axial direction so that the blades 142 may be shaped to produce a greater amount of work on the air and accordingly increase the pressure ratio to a greater extent for a given hub and flow path configuration.

While the above invention has been described in connection with a particular axial flow compressor and gas turbine engine configuration, it should be apparent to those skilled in the art that this compressor unit may be adapted for use with any number of engine configurations without departing from the spirit of the invention. Accordingly, the scope of the invention is to be determined solely by the appended claims.

What is desired to be secured by Letters Patent of the United States is:

1. Apparatus for pressurizing air in a gas turbine engine, said apparatus comprising:
an axial flow compressor rotating in a first direction for discharging air at an absolute velocity lower than the speed of sound;
a counter-rotating bladed impeller having a generally axial directed inlet portion defining a substantially undeflected flow path for receiving the air discharged from said axial compressor, the rotational speed of said bladed impeller being sufficient to produce a relative velocity between said discharge air and the impeller inlet portion greater than the speed of sound, thereby producing shock waves adjacent the inlet portion of said bladed impeller to reduce the absolute velocity of air flowing therethorugh below sonic and pressurize it;
said bladed impeller having an outlet portion defining a flow path radially outward from said inlet portion whereby air is substantially radially accelerated and diffused for further increase of its pressure.

2. Apparatus as in claim **1** wherein:
said bladed impeller comprises a hub defining the inner bounds of an annular flow path thereacross and a series of radially extending blades secured thereto, said blades having a configuration so that they define, in combination with said hub, flow paths having a relatively constant flow area in said inlet portion wherein said shock waves are produced and a diverging flow area adjacent the outlet portion thereof for diffusion of air.

3. Apparatus as in claim **1** wherein:
said axial flow compressor is a transonic compressor having a minimum number of stages for pressurizing air.

4. Apparatus as in claim **2** wherein:
said impeller comprises a mixed-flow impeller having a generally axially directed inlet portion and generally outwardly directed outlet portion;
said axial compressor imparts a substantial tangential velocity component to said air for discharge to said mixed flow impeller;
the chord of said impeller blades in their inlet portion is angled towards a tangential direction for generally parallel relative flow of air from said axial flow compressor into the inlet of said impeller;
the chord of said impeller blades adjacent the inlet portion has a minimum curvature along the portion wherein said shock waves are produced;
the chord of said impeller blades downstream of the inlet portion wherein said shock waves are produced is curved to a generally axial direction at the discharge end of said impeller.

5. Apparatus as in claim **4** wherein:
the flow path through said impeller blades is turned radially outward in its outlet portion with an angle substantially less than 90 degrees relative to the axis of said impeller;
said mixed-flow impeller is rotated at a speed sufficiently great so that a substantial amount of the pressurization of the air passing through said impeller is produced by the shock waves at the inlet thereto whereby the radial distance between the inlet and the outlet portions of said impeller is minimized.

6. Apparatus as in claim **4** further comprising:
an annular diffuser positioned to receive the discharge from said impeller and having a diverging flow area whereby the pressure of the air discharged from the mixed flow impeller is further increased.

7. Apparatus as in claim **6** in combination with a gas turbine engine, said gas turbine engine comprising:
a combustor means positioned to receive pressurized air from said diffuser, said combustor means including means for mixing said pressurized air with fuel and igniting the mixture to produce a relatively hot gas stream;
and turbine means positioned downward of said combustor means for extracting a portion of the energy from said hot gas stream to drive said axial flow compressor and said mixed flow impeller.

8. Apparatus as in claim **7** wherein said turbine means comprises:
an annular inlet nozzle having a series of radial vanes positioned downstream of said combustor means to receive and accelerate said hot gas stream;
a first stage of turbine blades positioned downstream of said inlet nozzle for rotation in response to the passage of said hot gas stream and a shaft extending from said first stage to said mixed flow compressor for corotation;
a second stage of turbine blades positioned downstream of said first stage for rotation in response to passage of said hot gas stream;
shaft means coaxial with said first-mentioned shaft means and connecting said axial flow compressor and said second turbine stage for corotation.

9. Apparatus in claim **8** wherein said combustor means comprises:
an annular chamber having the inlet from said diffuser positioned in a radially outward portion thereof and an outlet portion positioned radially inward of and adjacent said inlet portion;
at least one burner unit positioned in said chamber for mixing pressurized air with fuel and sustaining combustion thereof, said burner unit being positioned to direct the hot gas stream produced by combustion in a forward direction relative to the flow through said compressor unit and said turbine means;
duct means for receiving hot gas stream from said burner unit and turning the flow for discharge into said annular turbine nozzle, thereby minimizing the axial length of said engine.

10. Apparatus as in claim **9** wherein said engine further comprises:
power turbine means positioned downstream of said turbine means for extracting a major portion of energy from said hot gas stream to produce a mechanical torque output.

11. Apparatus as in claim **1** wherein said mixed flow blades impeller comprises:
a rotatable hub defining the inner bounds of an annular flow path across said impeller;
a plurality of generally radially extending blades secured to said hub and defining a flow path for the inlet portion of said impeller;
a plurality of generally radially extending blades secured to said hub and spaced from said inlet vanes for forming a flow path through the outlet portion of said impeller;
said apparatus further comprises:
a generally annular housing having an inner surface closely adjacent the outer ends of said vanes for defining the outer bounds of the flow path across said impeller;
a series of radially extending stator vanes secured to

said housing and extending towards said hub between said vanes;

said stator vanes being shaped to turn the air passing thereacross so that the outlet vanes produce a maximum pressurization of said air.

12. Apparatus as in claim **1** wherein said axial flow compressor and said counterrotating blade impeller are operated to produce relatively high Mach numbers at the entrance to said blade impeller, and wherein said apparatus further comprises:

a plurality of stator vanes interposed between the downstream end of the axial flow compressor and the inlet of the counterrotating blade impeller for receiving air from the axial flow compressor and directing it to the inlet of the blade impeller, said stator vanes being shaped to produce a minimum turning to the flow discharged from the axial flow compressor for the radially inward portion of the vanes and to produce a substantial turning of the flow discharged from the radially outward portion of the axial flow compressor over the radially outward portion of the stator vanes for minimizing radial variations of relative Mach numbers at the inlet to the blade impeller.

13. Apparatus as in claim **12** wherein said stator vanes are shaped at their radially outward portion to produce a substantial turning of the air flow from a tangential direction to an axial direction relative to the axis of the bladed impeller.

14. Apparatus as in claim **13** wherein the radially inward portion of the vanes is shaped to turn the flow discharged from the inner portion of the axial flow compressor towards a tangential direction.

References Cited

UNITED STATES PATENTS

2,689,681	9/1954	Sabatiuk	230—123
2,842,306	7/1958	Buchi	230—119X
3,037,349	6/1962	Gassmann	60—39.16

CARLTON R. CROYLE, Primary Examiner

U.S. Cl. X.R.

417—205, 247