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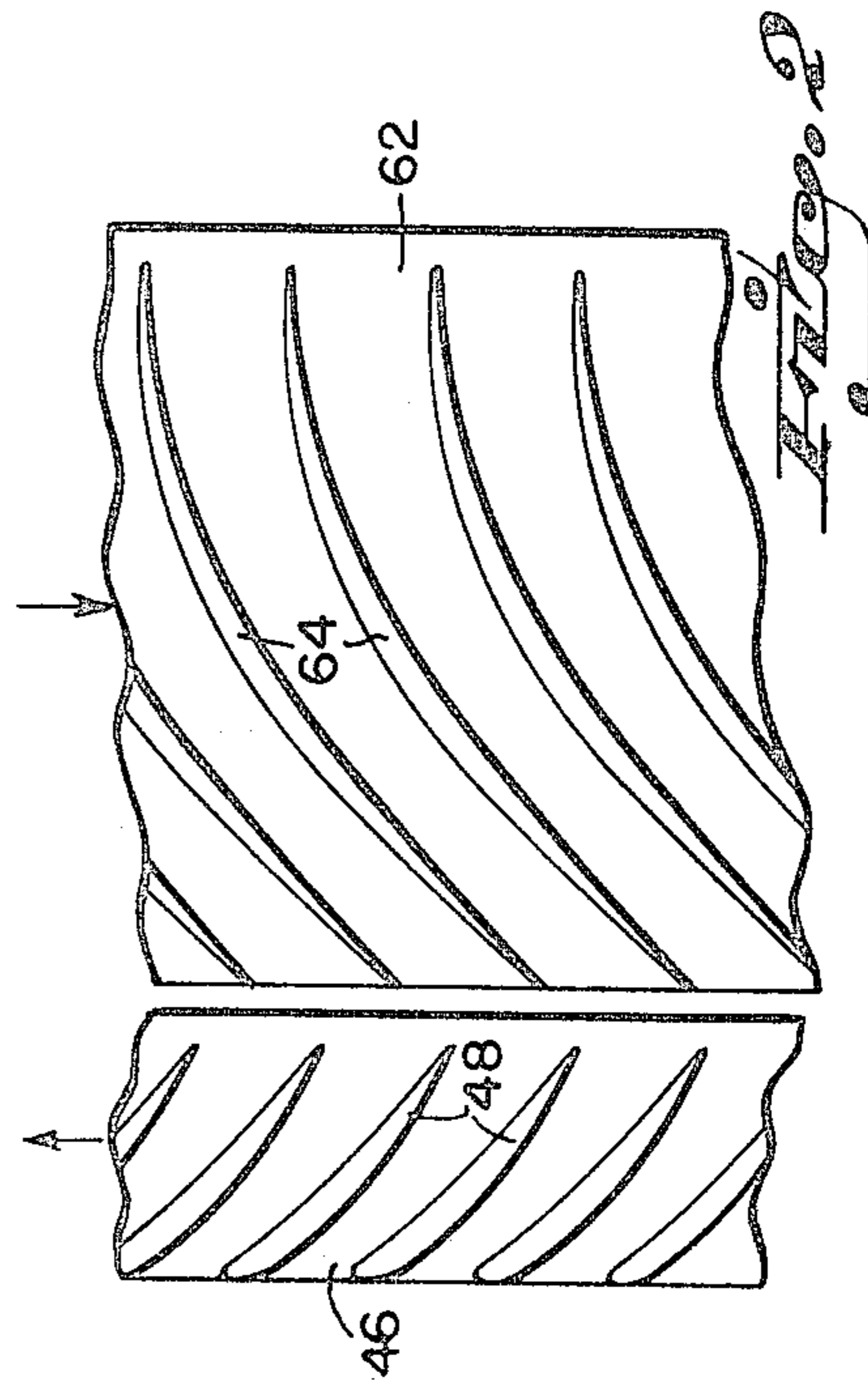
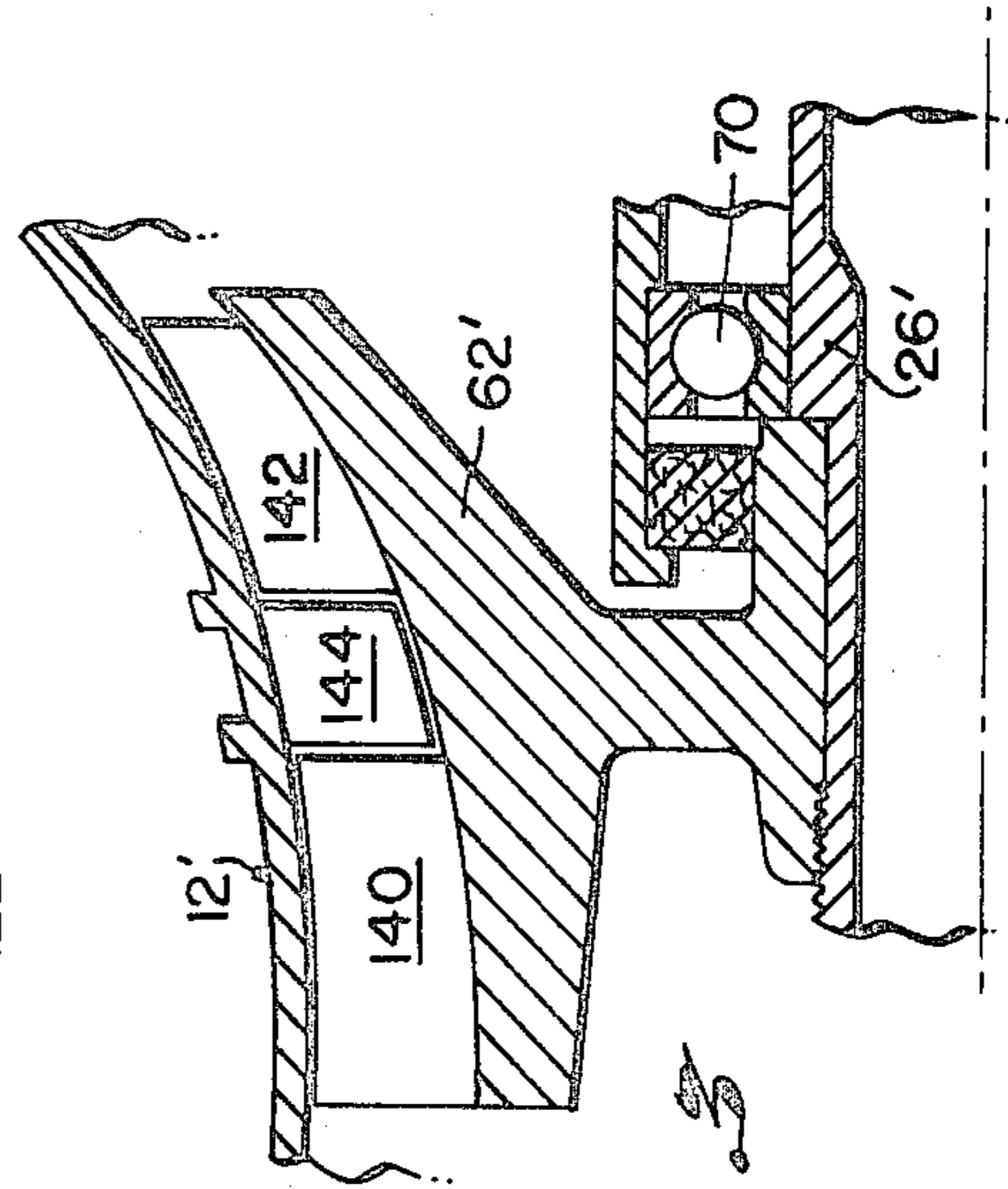
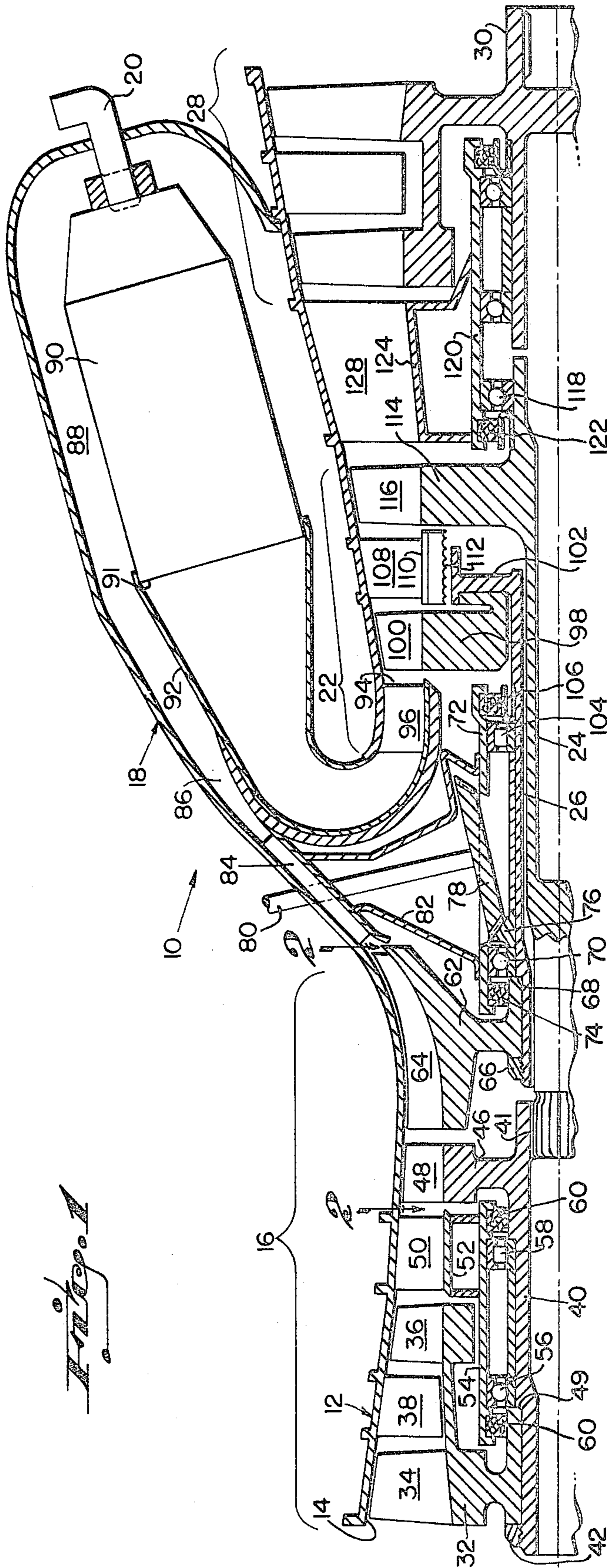
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3,546,880

COMPRESSORS FOR GAS TURBINE ENGINES

Filed Aug. 4, 1969

3 Sheets-Sheet 1



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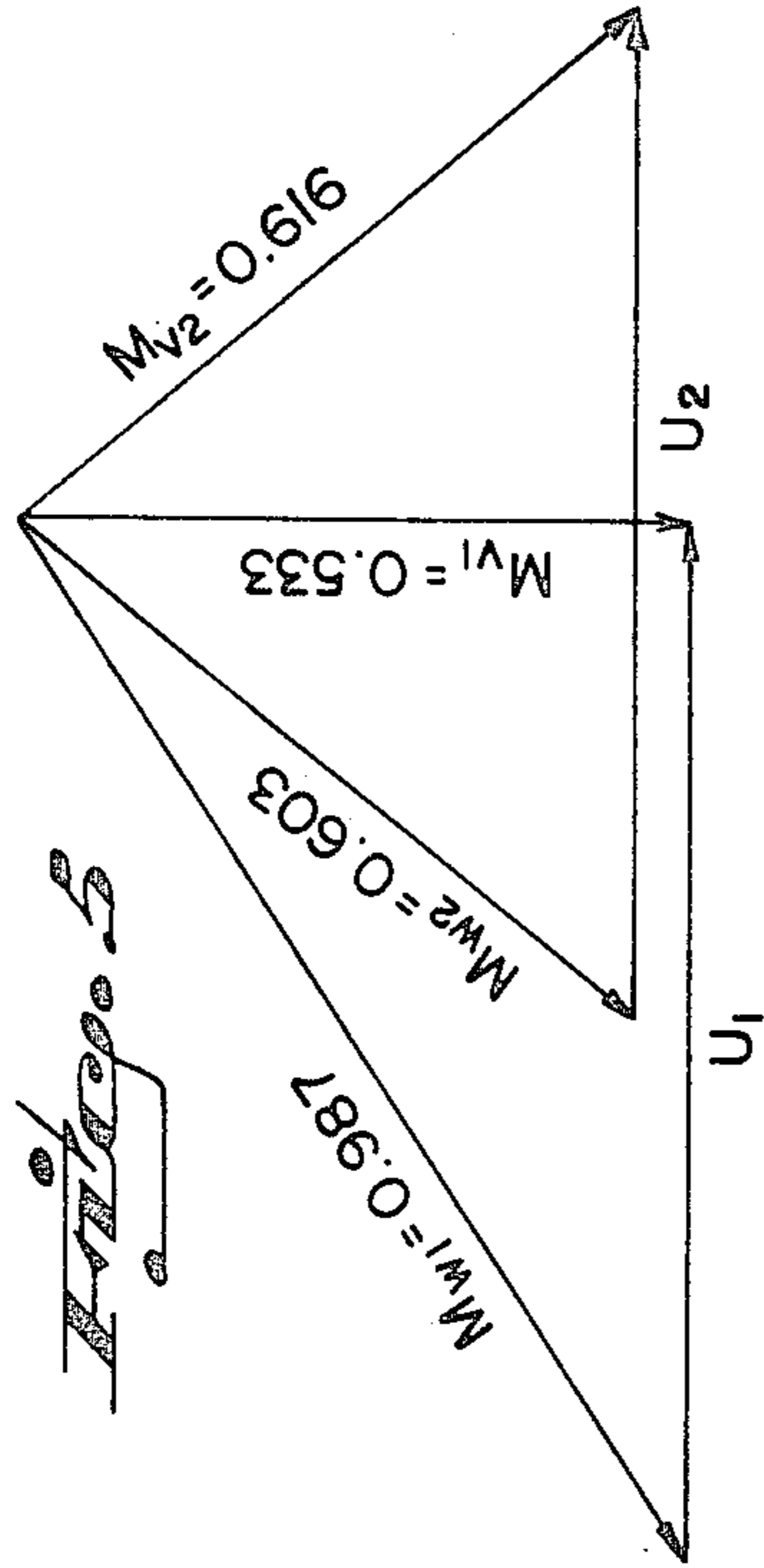
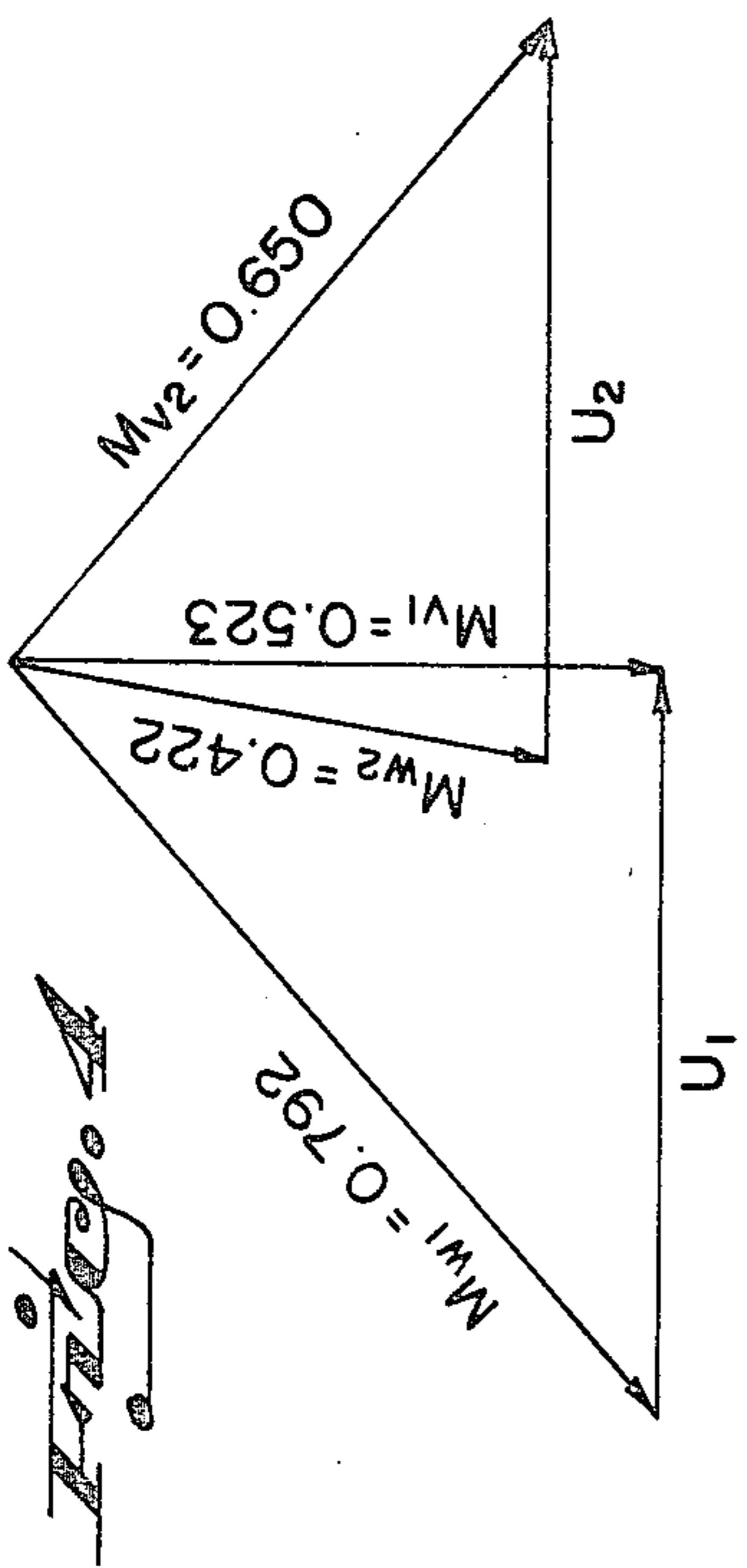
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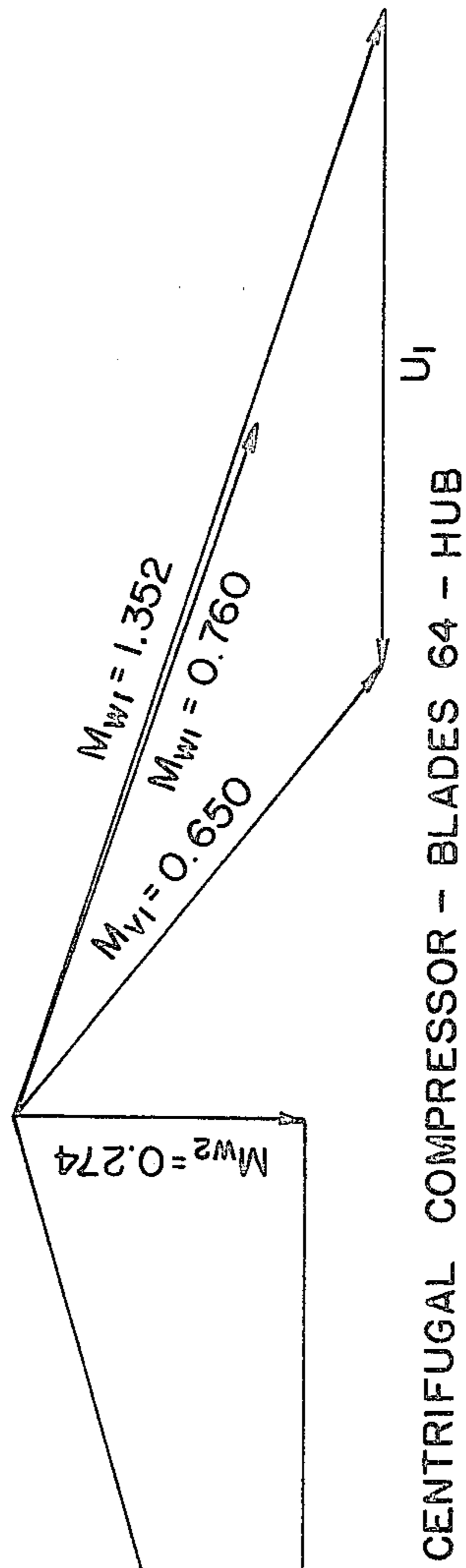
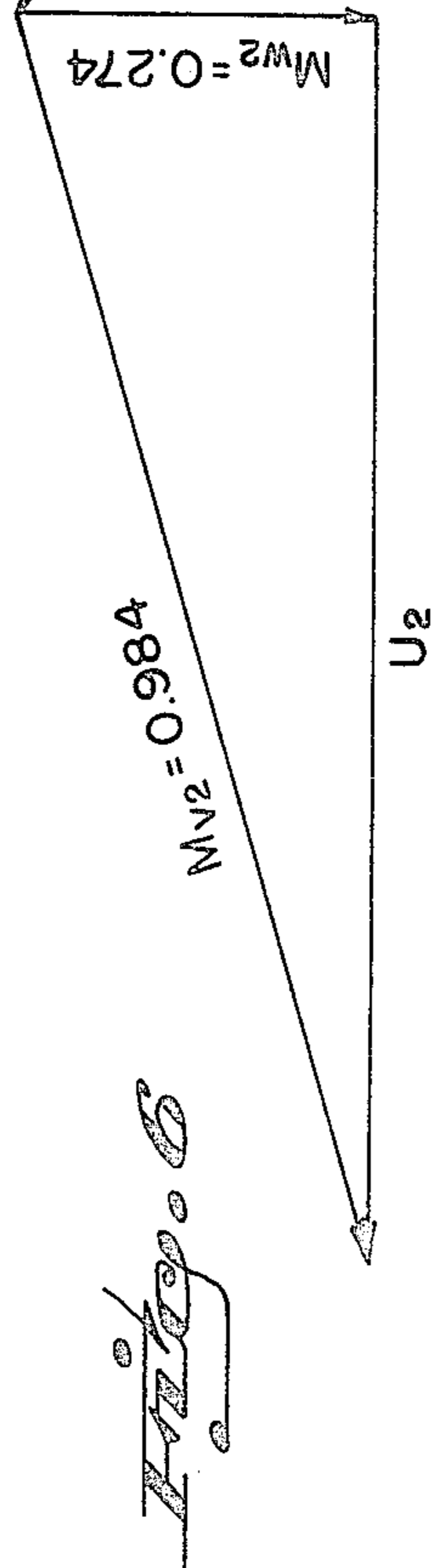
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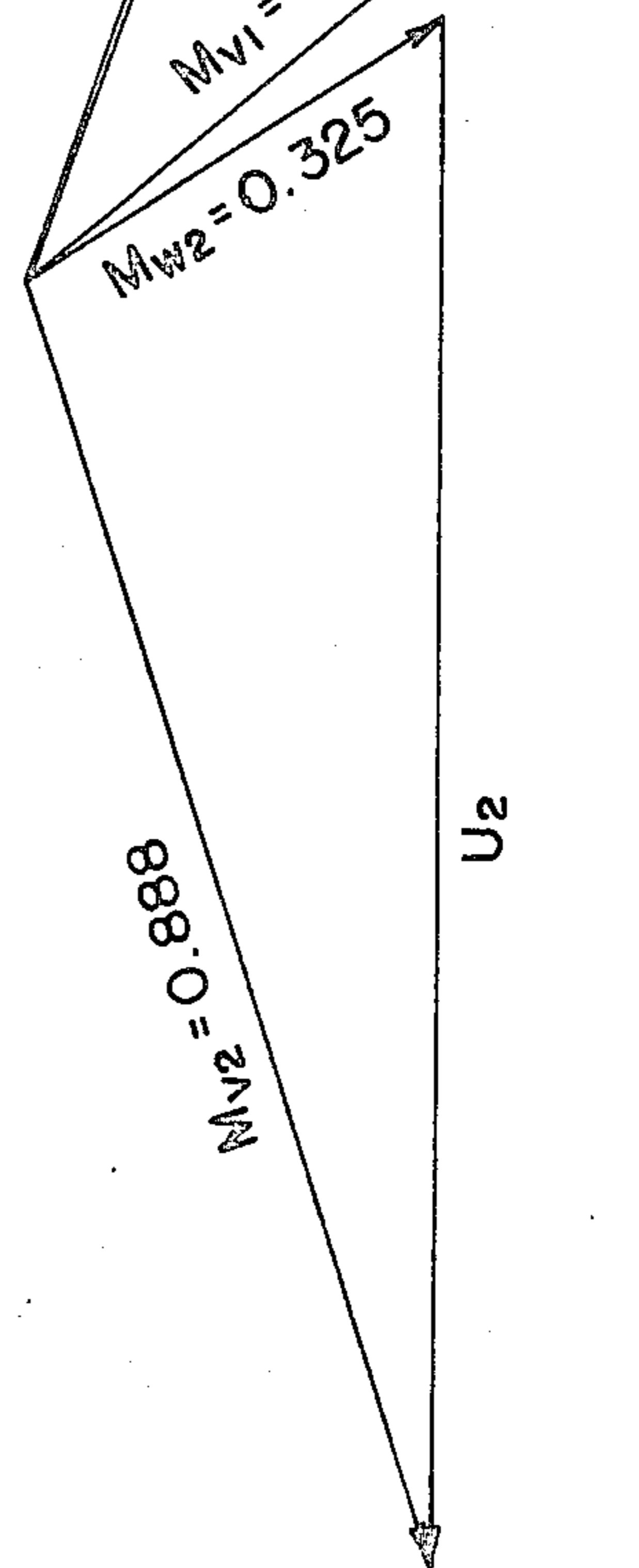
AXIAL COMPRESSOR - BLADES 48 - HUB

AXIAL COMPRESSOR - BLADES 48 - TIP



CENTRIFUGAL COMPRESSOR - BLADES 64 - HUB

CENTRIFUGAL COMPRESSOR - BLADES 64 - TIP



CENTRIFUGAL COMPRESSOR - BLADES 64 - TIP

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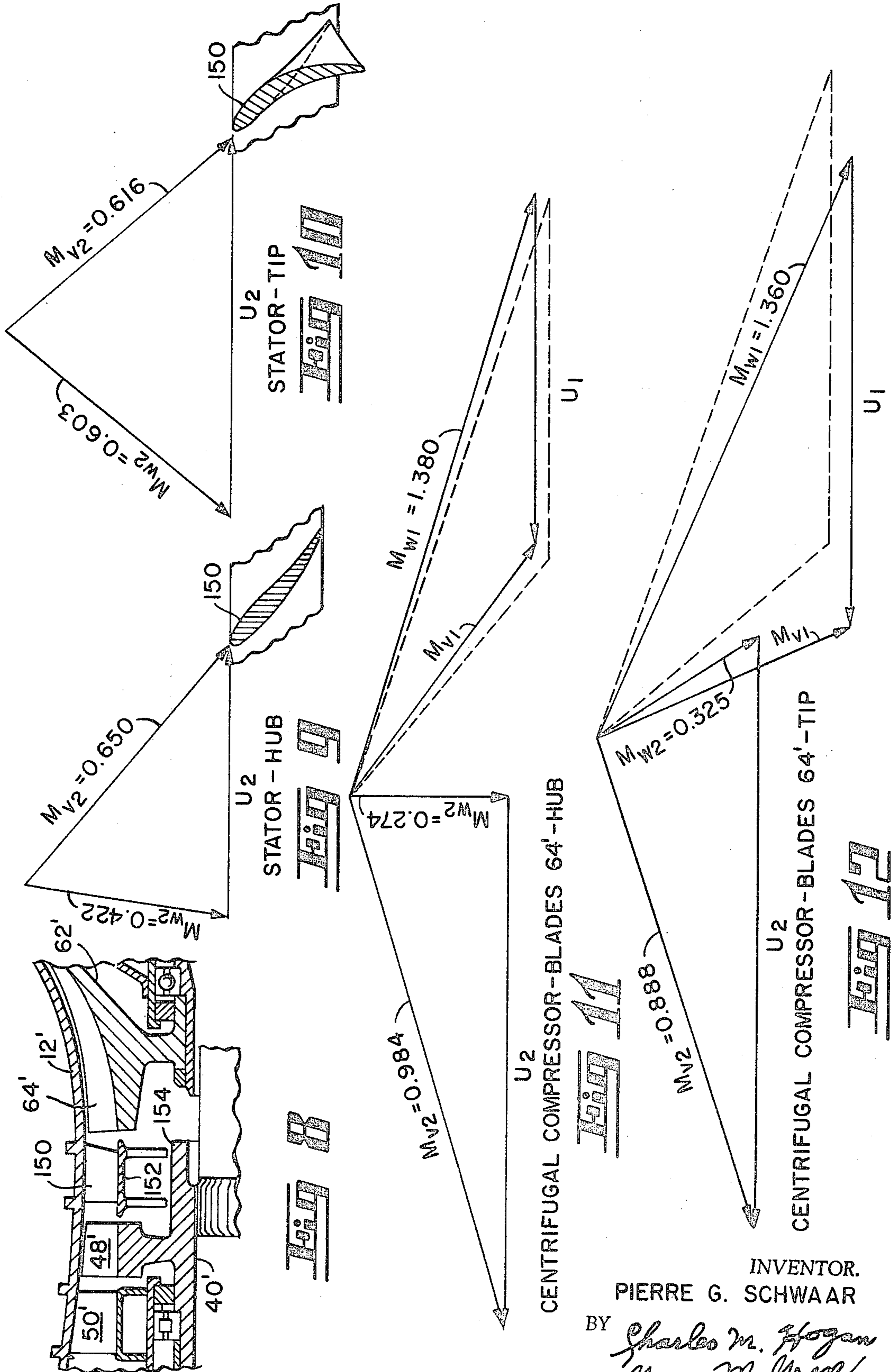
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COMPRESSORS FOR GAS TURBINE ENGINES

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3 Sheets-Sheet 3



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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 control, 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

2

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 labyrinth 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 blading, 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 mem-

ber 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 adapted 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

7

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 therethrough 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 im-

8

PELLER 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.

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U.S. Cl. X.R.

417—205, 247