May 9, 1961 P. PISA 2,983,434

TURBOCOMPRESSOR

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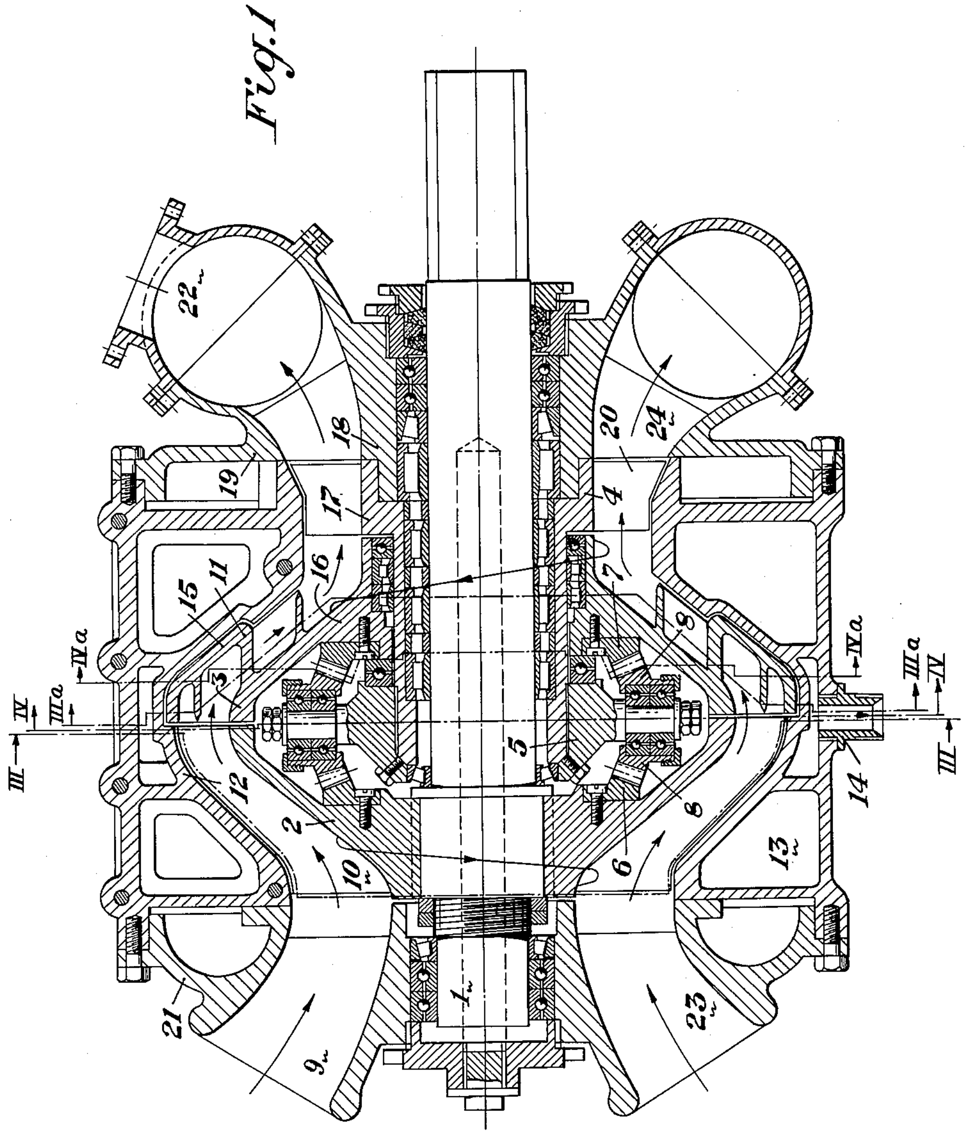

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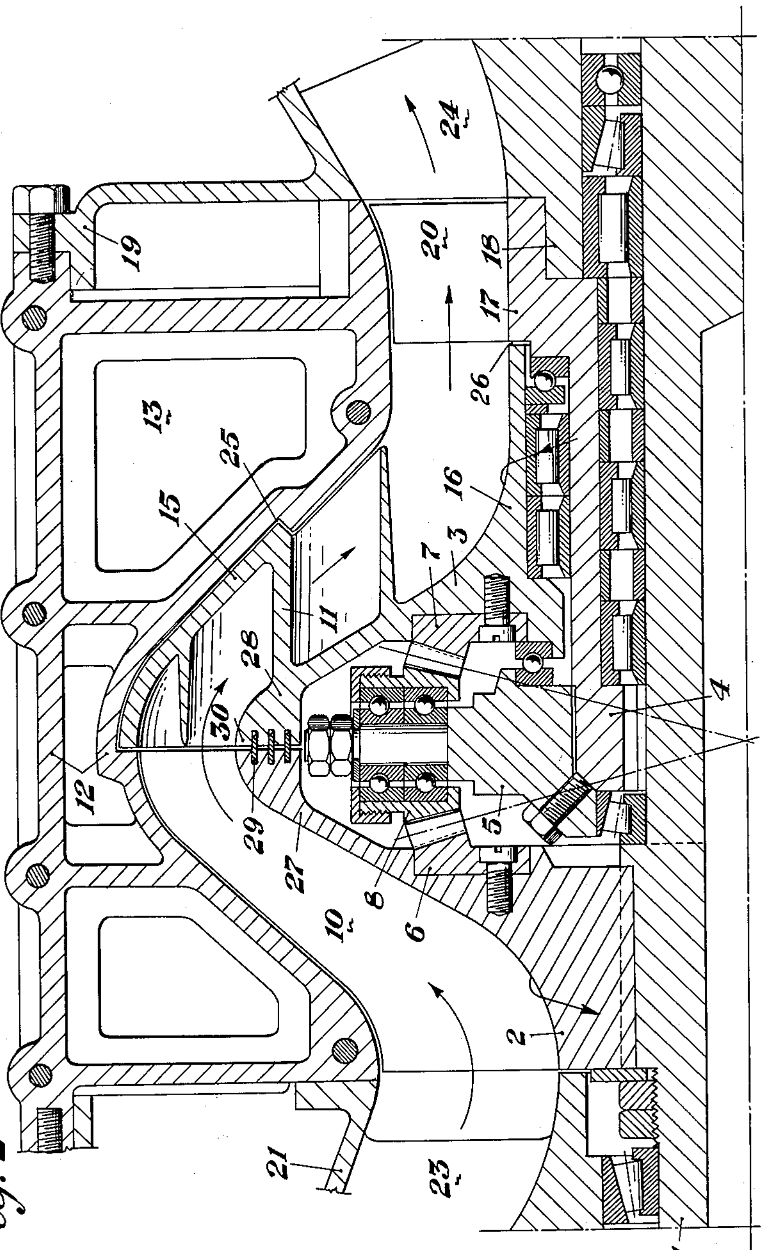

Fig. 2

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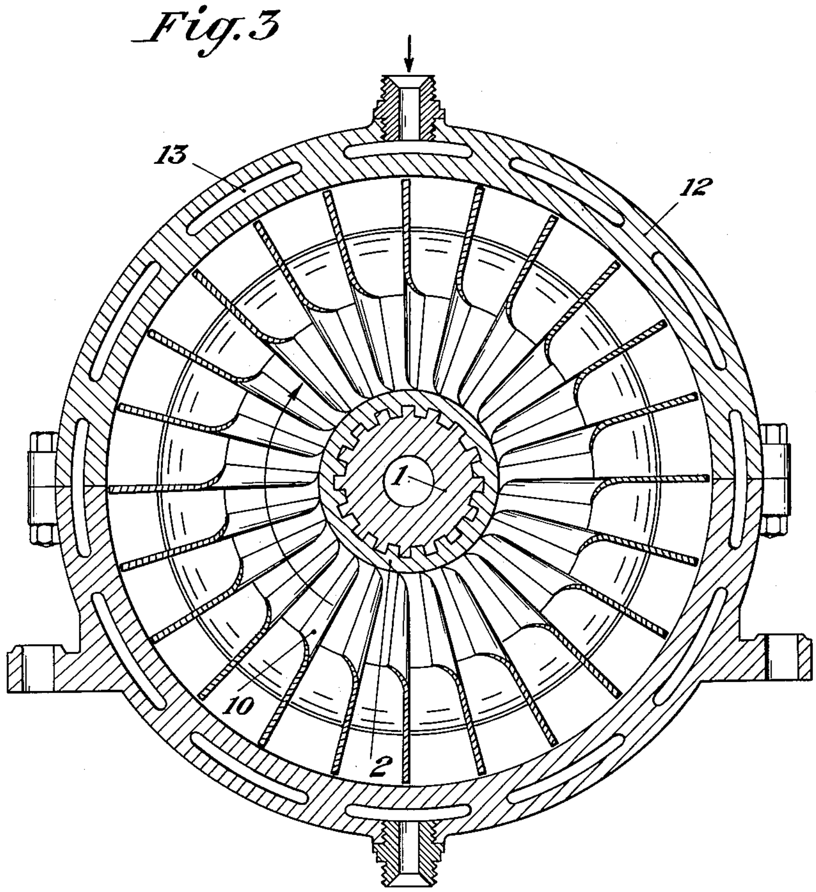
Fig. 3



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TURBOCOMPRESSOR

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Fig. 3a

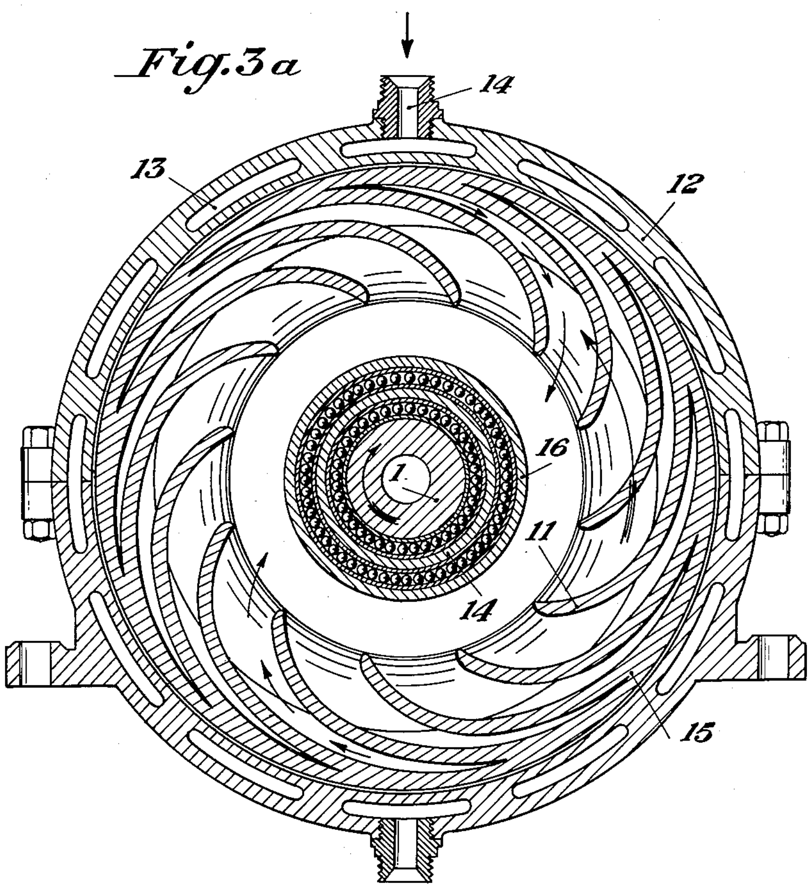

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Fig. 4

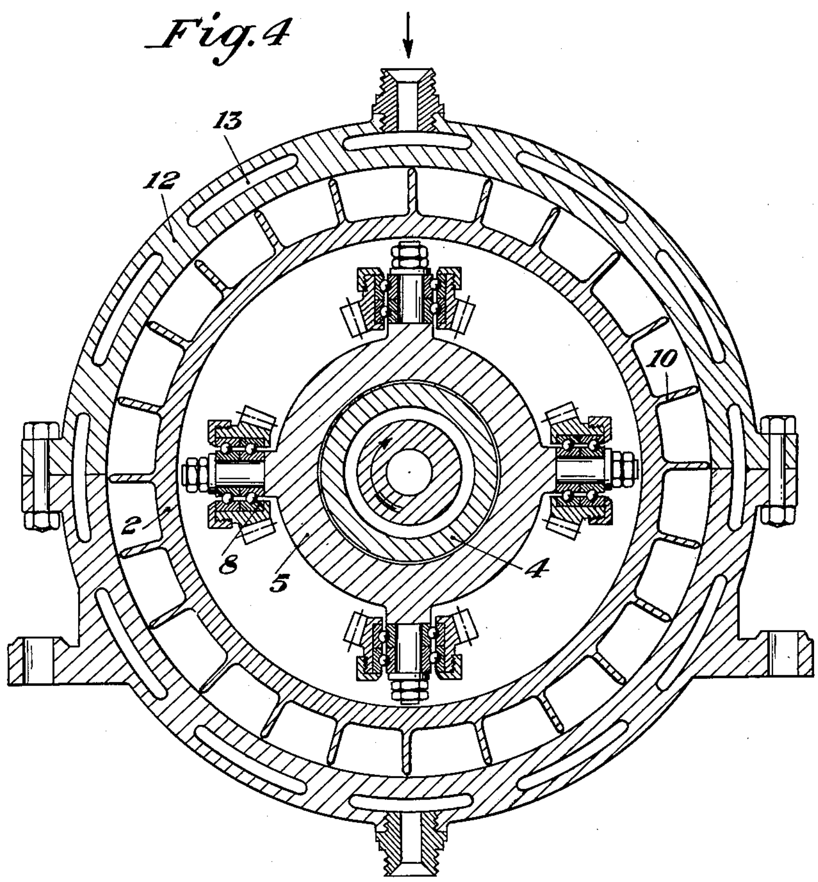

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Fig. 4a

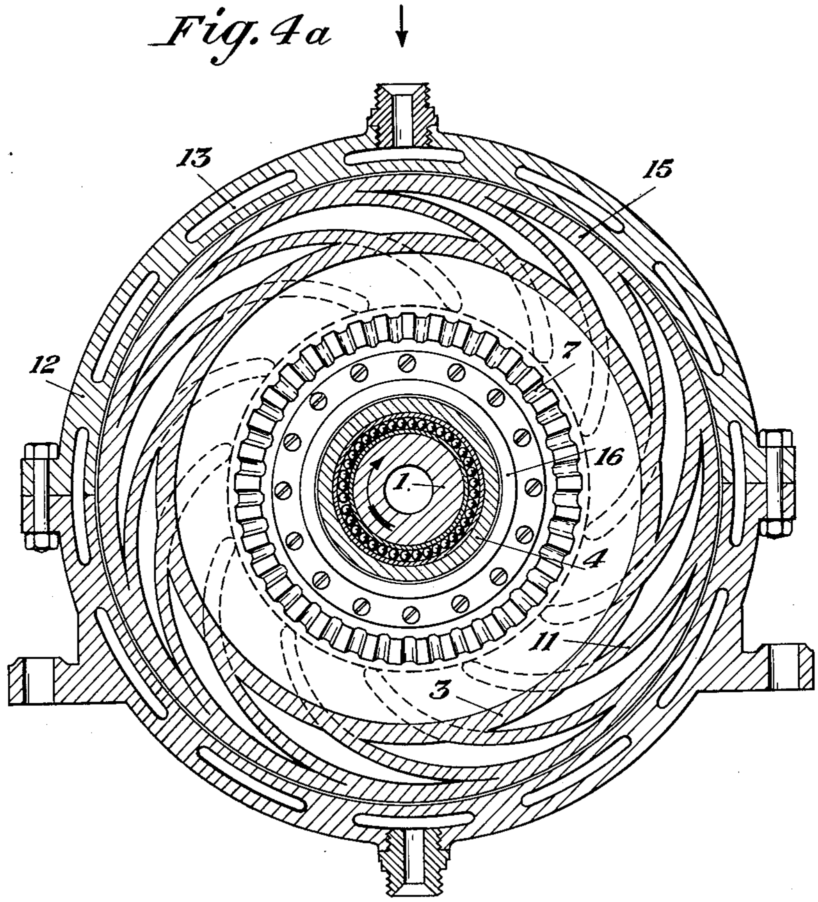

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Fig. 5

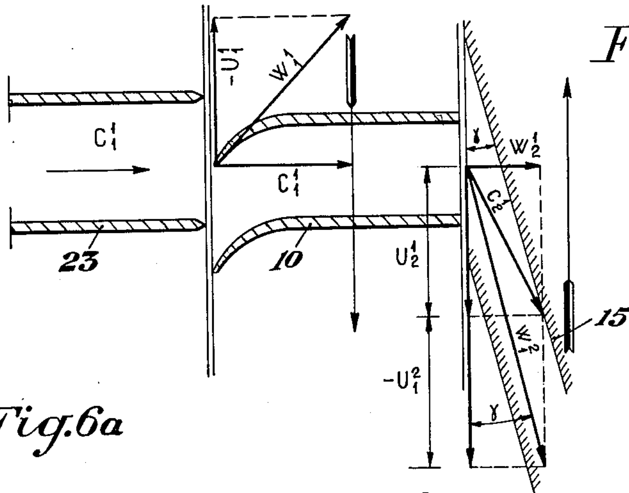

Fig. 6a

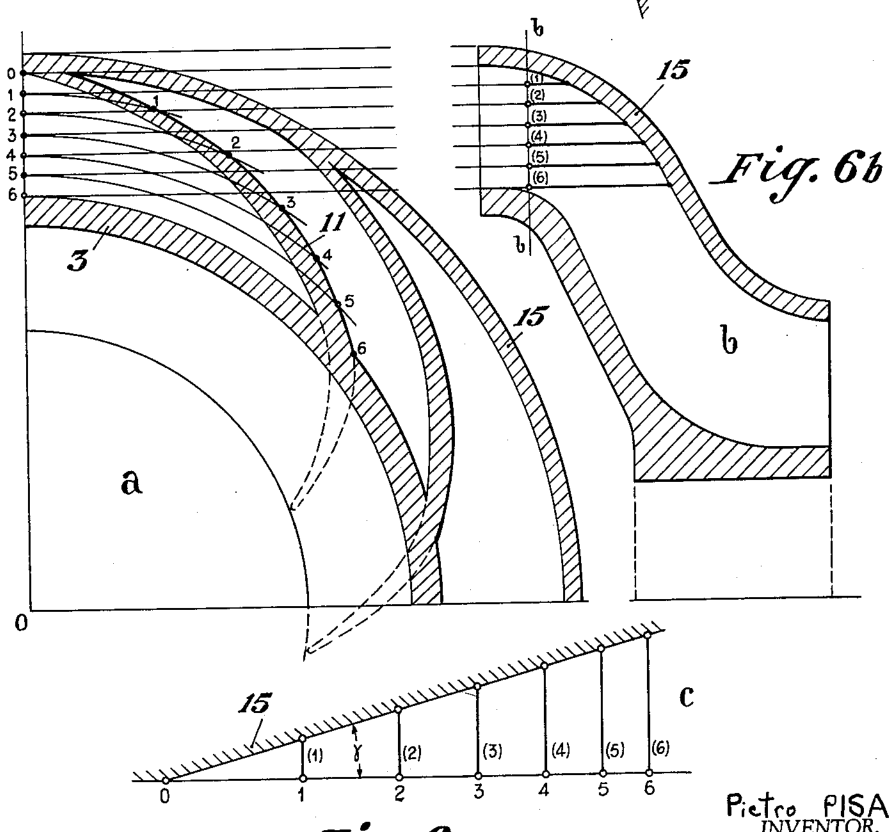

Fig. 6b

Fig. 6c

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2,983,434
Patented May 9, 1961

2,983,434

TURBOCOMPRESSOR

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Filed Dec. 24, 1957, Ser. No. 704,975
5 Claims. (Cl. 230—124)

The present invention relates to a turbocompressor of centrifugal type, wherein the diffuser is movable instead of being stationary, said diffuser revoluting in reverse direction with respect to the centrifugal impeller, with the purpose of obtaining a high pressure for the air by its passage through said two subsequent elements forming one stage of the turbocompressor.

The relative inflow speed, or air flow speed through the ducts, is therefore the sum of the high peripheral speeds of the centrifugal impeller and of the diffuser rotating in the reverse direction, while a low outflow speed is obtained and therefore a great amount of kinetic energy is converted to pressure.

The air path through the turbocompressor is part-way radial and part-way axial, and the shape of the machine is somewhat similar to that of a centrifugal turbocompressor.

A feature of the contra-rotating diffuser resides in that it keeps unaltered the efficiency when the number of revolutions changes, in that the air moves always therethrough without striking the blades and the flow defining walls. The machine, in one simple embodiment given only by way of example is shown in the attached drawings, wherein:

Fig. 1 is an axial cross-sectional view of the turbocompressor according to the invention;

Fig. 2 shows in enlarged scale the details of the connection between the centrifugal impeller and the contra-rotating diffuser;

Fig. 3 is a cross-sectional view taken along line III—III of Fig. 1;

Fig. 3*a* is a cross-sectional view taken along line III*a*—III*a* of Fig. 1;

Fig. 4 is a cross-sectional view taken along line IV—IV of Fig. 1;

Fig. 4*a* is a cross-sectional view taken along line IV*a*—IV*a* of Fig. 1.

Fig. 5 shows graphs or velocity triangles formed by the velocities of the impeller and the diffuser and the particular velocity of flow of the air through the ducts;

Figs. 6*a*, 6*b* and 6*c* show the shape of the bottom walls of the contra-rotating diffuser.

The shaft 1, is actuated from the end of the machine corresponding to the outlet manifold; said shaft is supported at its ends and in an intermediate zone (Fig. 1).

A centrifugal impeller 2 is keyed to said driving shaft, while the contra-rotating diffuser 3, facing the impeller and co-axial with said impeller, rotates on a sleeve 4 which is held stationary in that it is rigidly connected to the compressor shell (Figs. 1 and 2).

Both the impeller 2 and the contra-rotating diffuser are provided with the gear rings 6 and 7 which engage the sprocket 8 mounted by the stationary spider 5 screw threaded to the sleeve 4 (Figs. 1 and 2).

The peripheral velocities of the impeller 2 and of the contra-rotating diffuser are therefore equal and directed in contrary directions, while part of the driving moment

is transmitted from the impeller to the facing diffuser (Figs. 1, 2, 4 and 4*a*).

The air which comes from the intake 9, and moves in a mainly axial direction, is first radially deflected through the ducts of the centrifugal impeller 2 and its axial flow is then restored at the outlet from said impeller (Figs. 1 and 2).

The direction changes are obtained in the impeller with great radii of curvature.

The air enters the contra-rotating diffuser in axial direction, then the air stream is deflected by the diffuser blades toward the rotation axis, and finally an axial flow direction is restored at the outlet from the machine (Figs. 1 and 2). The path of the ducts of the impeller 2 is very similar to that found in the centrifugal compressors where the blades 10 have a radial direction (Figs. 1 and 3).

The air stream through the contra-rotating diffuser has however a different shape, in that said stream progressively comes nearer to the center O of the diffuser while moving through ducts defined by spiral shaped blades 11 (Fig. 3*a*). These ducts have at their outer portion a narrow cross-sectional inlet area (almost tangential inlet), said area continuously increasing while nearing to the center due to change of direction (almost radial outflow—Fig. 3*a*).

The progressive increase of the cross-sectional passage area causes a slowing down of the air stream and therefore an increase of the pressure. The inflow velocity into the diffuser ducts is high, for being equal to the sum of the two equal and contrary peripheral velocities of the impeller and of the diffuser, respectively, while the outflow velocity is low, owing to the remarkable increase of the passage cross-sectional area (Figs. 1 and 2).

The air stream moving through the impeller 2 and the last portion of the contra-rotating diffuser 3 directly contacts the stationary walls of the case 12 within which glides, through suitable jackets 13, the cooling water, the inlet and the outlet of which are formed by the fittings 14 (Figs. 1, 3 and 3*a*).

The blades 10 of the impeller 2 have no outer connecting wall (open ducts) while, for constructional and sturdiness reasons, the blades 11 of the contra-rotating diffuser 3, near to the inlet zone are bodily connected to a cone shaped connecting wall or outer shroud ring 15 (Figs. 1, 2 and 3).

These two movable members are sufficiently compact to withstand both the stresses of the centrifugal force and the moment or stress that said members must impart to the air stream (blade pressure). The axes of both the impeller 2 and the contra-rotating diffuser 3 are coincident; it is, however, to be noted that these two members are co-axial but do not rotate above one another (Figs. 1 and 2).

In fact, the sleeve 4, by the intervention of roller bearings, supports the intermediate portion of the driving shaft 1 received therein, while the outer surface of said sleeve supports the hollow shaft 16 of the diffuser, also in this case by the intervention of roller bearings (Figs. 1 and 2).

The sleeve 4, which carries at its free end the spider 5 carrying the intermediate bevel pinions 8, must be steadily located and it must withstand the stress imposed to the pinions 8 (twist moment). This stress is due to the driving moment applied by the impeller 2 to the contra-rotating diffuser 3 by means of the respective bevel gear rings, 6 and 7, respectively (Figs. 1, 2 and 4*a*).

The sleeve is provided, at its end remote from the impeller, with a great ring 17 which is supported from inside by the hub 18 rigid with the rear cover 19. The shaped blades 20, rigid or integral with said ring 17, bear against the inner walls of the shell **12** of the turbocompressor (Figs. 1 and 2).

The shell **12** is comprised of two halves and may be opened along a horizontal plane (Figs. 1, 3, 4 and 4*a*). Both the fore cover, carrying the intake **9**, and the rear cover **19**, carrying the outlet manifold **22**, are fastened to the shell **12**, when the latter has been closed, and said two covers, by means of the cast vanes **23** and **24** are rigidly connected to the supporting hubs for the driving shaft **1**, at its ends, also in this case by the interposition of ball bearings and thrust bearings for adjusting purposes, as shown in Figs. 1 and 2. The operation of the turbocompressor is evident from the velocity triangles of the air stream and of the movable members (Fig. 5).

The air reaches the impeller from the intake with an absolute inflow velocity C_1^1 which, when composed with the inner peripheral velocity U_1^1 gives, as resultant, the relative inflow velocity W_1^1 (Fig. 5).

The stream is moved, by the centrifugal force through the conduits of the impeller **2** defined between the blades **10** which have a perfectly radial direction. The air outflows in axial direction, the deflection of the stream being obtained by the curvature of the walls of the shell **12** and of the impeller **2** (Figs. 1 and 2). The relative outflow velocity W_2^1 is less than the outer peripheral velocity U_2^1, and than the relative inflow velocity W_1^1, as readily seen from Fig. 5.

The air pressure increases due to the effect of the centrifugal force (proportional to the difference of the squares of the velocities) and due to the slowing down of the flow speed through the ducts of the impeller.

The stream leaves the impeller **2** with an outflow absolute velocity C_2^1, which is the resultant of the outflow relative velocity W_2^1 and of the outer peripheral velocity U_2^1 of the impeller, as shown in Fig. 5. The velocity C_2^1 has a certain inclination with respect to the inlet plane of the diffuser and passes through said plane. (The peripheral velocities U_1^1, as well as the relative velocities and the absolute ones, are calculated on the base of the medium radius of the inlet and outlet cross-sectional areas of both the impeller and the diffuser).

As the contra-rotating diffuser **3** has a peripheral velocity U_1^2 equal and contrary to the velocity U_2^1 of the impeller **2**, the relative inflow velocity W_1^2 in the diffuser is given by the resultant of the absolute outflow velocity C_2^1 from the impeller **2** and of the peripheral $-U_1^2$ (with the inverted sign) proper of the contra-rotating diffuser **3**, as shown in Fig. 5.

Therefore the inflow relative velocity W_1^2 into the diffuser will have a very remarkable value, as it is to be noted from Fig. 5, said value being the sum of the peripheral velocities of the impeller and of the diffuser.

The relative inflow velocity W_1^2 will therefore have a small inclination (angle γ—Fig. 5) with respect to the rotation plane of the inlet cross-sectional area of the diffuser.

Fig. 6 shows as by a graphical method has been found the inclination γ with respect to said plane that the corner or common line between the convex wall (back wall) of the blades **11** makes with the bottom conical wall of the shroud ring **15** of the diffuser (Figs. 1 and 2). By the projections method for six points of the blade backside, as illustrated in Fig. 6*a*, said points being decreasingly spaced apart from the center O of rotation, the corresponding axial depths have been found, as shown in Fig. 6 in thickened lines, for the ducts defined by the spiral shaped blades and the bottom conical wall of the diffuser.

These segments begin at the right-hand zone of the line *b—b*, as it will be seen in Fig. 6*b*, while in the right-hand zone there is the inlet zone of the diffuser, where the blades **11** present to the air stream a very sharpened edge, in order to avoid any impact in the inlet stream (Figs. 1 and 2). Taking the distances **0–1**, **0–2** and so on, as abscissae and the corresponding segments **1**, **2** and so on, as ordinates, the line or corner common to both said walls has been plotted as shown in Fig. 6*c*, said corner presenting with respect to the rotational plane the same angle γ that makes the inflow relative velocity W_1^2 when entering the diffuser. This line or corner represents the profile of all the conical bottom wall of the diffuser which is therefore tangent at any point to the inflow relative velocity W_1^2.

Having thus avoided any shock of the stream against the bottom wall of the diffuser, the air is progressively deflected towards the rotation center O by the concave wall of the spiral shaped blades **11**, as shown in Figs. 2 and 3*a*. The air issuing from the impeller due to both its own velocity and the velocity of the diffuser moving towards said air, is very quickly picked up by the diffuser.

Through the diffuser ducts, having a progressively increasing cross-sectional area, the air flow velocity is progressively slowed down, and therefore a remarkable conversion of kinetic energy into pressure occurs, which is proportional to the differential between the squares of the inflow and outflow relative velocities W_1^2 and W_2^2 respectively.

In the contra-rotating diffuser, the return stream, directed from down upstream, i.e. the air amount tending to escape along the edge **25** (Fig. 2) is held by the extended outer wall of the conical ring **15** which glides with a small clearance against the wall of the shell **12**.

At the end of the hub **16** of the contra-rotating diffuser **3**, along the flat wall faced to that of the stationary sleeve **4**, i.e. at the portion **26** (Fig. 2), the necessary clearance originates an air leakage, said air tending to move in reverse direction through the passage between the hub **16** of the diffuser, and the outer cylindrical surface, on which the diffuser rotates (Fig. 2). The air enters, therefore the space comprised between the conical disc **27** of the impeller and the inner disc **28** of the diffuser, best shown in Fig. 2, within said space being received the bevel ring gears **6** and **7**, as well as the pinions **8** (Figs. 1 and 2).

In order to reduce this leakage, a labyrinth seal is carried by the flat edges of said two discs, said seal being formed by the insert rings **29** projecting from the disc **27** and entering the registering grooves provided in the faced disc **28**, as clearly shown in Fig. 2.

These rings are made of antifriction material, so that they originate a little friction resistance and are easily replaceable.

For concision, the turbocompressor according to the present invention, has been disclosed with a single compression stage, while it must comprise at least another subsequent stage, in order to obtain that high compression feature which is the main feature of this great delivery compressor.

In this case, between the first and the second stage, an intermediate cooling stage must be provided, obtained by extended walls which are cooled from outside by means of a water circulation.

Another feature of the turbocompressor according to the present invention resides in that this machine keeps almost unchanged its efficiency, i.e. this turbocompressor affords the elimination of any shock of the air stream against the blades and the diffuser walls when the number of revolutions per minute of the machine falls down, below the normal rating.

In fact, from the examination of the velocity triangles, it is evident that for a lower number of revolutions, the peripheral velocities of both the impeller and the diffuser equally decrease due to the existing connection, and therefore also the absolute and relative velocities of the air stream at the inlet and at the outlet of the diffuser, proportionally are changed.

The new velocity triangles, which are thus obtained, are similar to those obtained in the normal operation and therefore remains unaltered the inclination γ to be provided for the bottom wall of the contra-rotating diffuser.

At parity of obtained compression, the machine according to the present invention rotates at a lower number of revolutions per minute (about one half) of the number of revolutions necessary in the conventional turbocompressors according to the prior art, with an obvious advantage due to the reduced stresses produced by the centrifugal force. The turbocompressor according to the present invention, having the features of a high compression and a great delivery, besides having a minor number of stages or passages, in comparison with the prior art machines, has been disclosed in the attached drawings only by way of example.

What is claimed is as follows:

1. A turbocompressor of the centrifugal type, which comprises in combination a casing having an intake therein and an outlet therefrom, a driving shaft rotatably mounted in said casing for rotation around an axis of rotation, an impeller mounted on said driving shaft within said casing, a diffusor rotatably mounted for rotation around said driving shaft and within said casing, gear means connected between said impeller and said diffusor for rotating said diffusor in a direction opposite to the direction of the rotation of the impeller and at the peripheral speed equal and opposite to the peripheral speed of said impeller, said impeller having a plurality of blades therein defining a first plurality of ducts each having an inlet and an outlet, and said diffusor having a plurality of blades thereon defining a second plurality of ducts each having an inlet and an outlet, the inlets to said first plurality of ducts being at the said intake of said casing and the outlets of said second plurality of ducts being at said outlet from said casing, and the outlets of said first plurality of ducts being opposed to the inlets to said second plurality of ducts, said first plurality of ducts extending away from the axis of rotation in a plane with the axis of rotation and an angle to the axis of rotation, said first plurality of ducts having a cross section gradually decreasing from the inlet toward the outlet of the ducts, and the ducts of said second plurality of ducts curving spirally inwardly toward the axis of rotation in the portions thereof adjacent to the outer periphery of said diffusor and curving toward a direction which is substantially parallel to the axis of rotation, said second plurality of ducts having a cross section which increases from the inlets toward the outlet, the inclination of the spiral portions being at a small angle to the plane of the face of the said diffusor in the initial section of the spiral and at the greatest inclination in the final section of the spiral toward the outlet of the ducts.

2. A turbocompressor of the centrifugal type, which comprises in combination a stationary casing, a front cover on the said casing and having an air intake therein, a rear cover on the said casing, an outlet manifold for the air attached to said cover, a stationary sleeve fastened at one end to the said casing and to the said rear cover and rigid with the rear cover and extending into said casing, a driving shaft rotatably mounted in the said sleeve for rotation around an axis of rotation, an impeller mounted on the said driving shaft within the said casing, a spider fastened to the portion of the said sleeve within said casing and spaced from said one end of the sleeve and leaving a free intermediate portion on the said sleeve, a sprocket rotatably mounted on the said spider, a diffusor rotatably mounted on the said intermediate free portion of said sleeve, said impeller comprising a disc and a plurality of blades defining a first plurality of ducts each having an inlet and an outlet, said diffusor comprising an inner disc, an outer disc and a plurality of blades therebetween which define a second plurality of ducts each having an inlet and an outlet, the disc of the said impeller and the inner disc of the said diffusor having edges which are opposed to each other and delimitate a chamber wherein the said spider is contained, a first gear ring fastened to the disc of the said impeller and engaging the said sprocket mounted on said spider, a second gear ring fastened to the inner disc of the said diffusor and engaging with the said sprocket mounted on the said spider for rotating said diffusor in a direction opposite to the direction of the rotation of the impeller and at a peripheral speed equal and opposite to the peripheral speed of said impeller, the second plurality of ducts of the said diffusor registering during rotation with the said first plurality of ducts of the said impeller and receiving from the said first plurality of ducts of the said impeller air having a speed equal to the sum of the peripheral speed of the said impeller and the said diffusor.

3. A turbocompressor according to claim 2, wherein the disc of the said impeller and the inner disc of the said diffusor have a labyrinth seal therebetween closing the chamber delimited by the last mentioned discs, said labyrinth seal comprising a plurality of rings fixed in the edge of the impeller disc, said inner diffusor disc having a plurality of grooves in the edge, said rings extending into the said grooves and sliding therein during rotation of said impeller and said diffusor and reducing the leakage therebetween.

4. A turbocompressor as claimed in claim 2, wherein said first plurality of ducts of the said impeller extend away from the axis of rotation in a plane with and at an angle to the axis of rotation, said first plurality of ducts having a cross section gradually decreasing from the inlet toward the outlet of the ducts, and the ducts of said second plurality of ducts in said diffusor curving spirally inwardly toward the axis of rotation in the portions thereof adjacent the outer periphery of said diffusor and curving toward a direction which is substantially parallel to the axis of rotation, said second plurality of ducts having a cross section which increases from the inlet toward the outlet, the inclination of the spiral portions being at a small angle to the plane of the face of the said diffusor in the initial section of the spiral and at the greatest inclination in the final section of the spiral toward the outlet of the ducts.

5. A turbocompressor as claimed in claim 4, wherein the spiral shaped blades of the diffusor have greatly tapered edges in the inlet zone to avoid shocks in the inflowing air stream, and have very sharp edges in the outlet zone to avoid eddy movements in the air stream leaving the ducts of the diffusor, the central portion of the said blades of the diffusor being thickened for rigid connection between the inner and outer discs of said diffusor.

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