Jan. 16, 1968 M. R. GARNIER 3,363,831

AXIAL-FLOW COMPRESSOR WITH TWO CONTRA-ROTATING ROTORS

Filed June 23, 1966 2 Sheets-Sheet 1

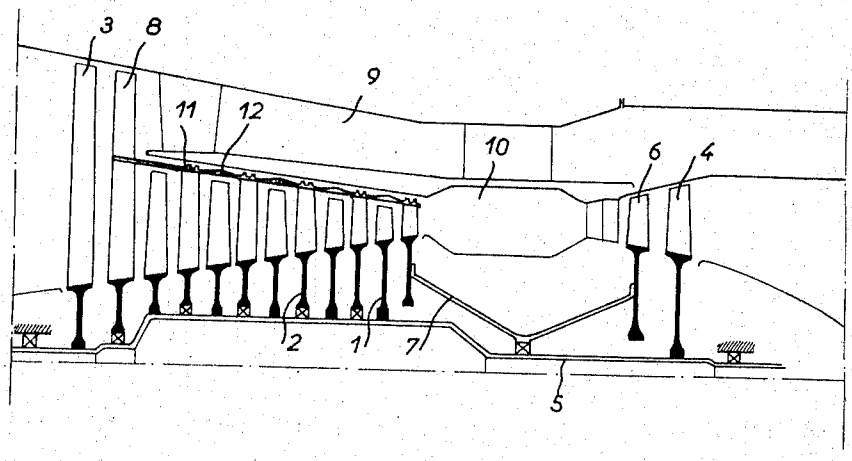

Fig. 1

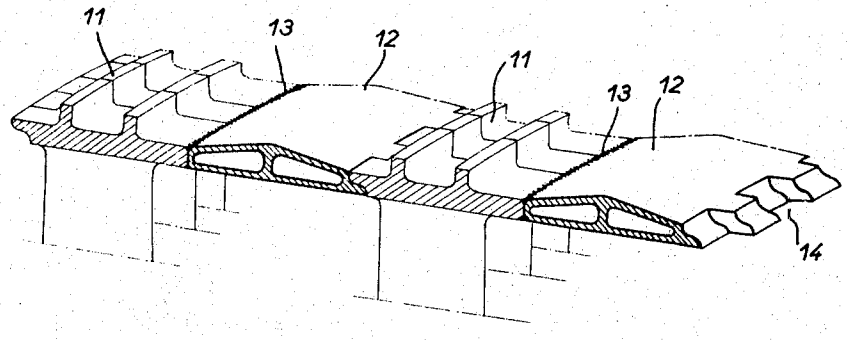

Fig. 2

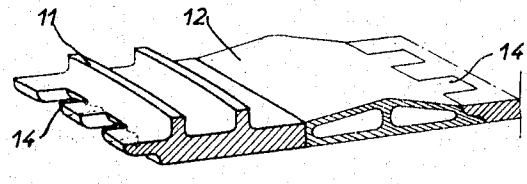

Fig. 3

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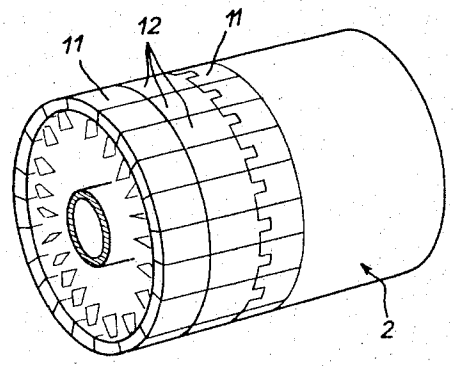

Fig. 4

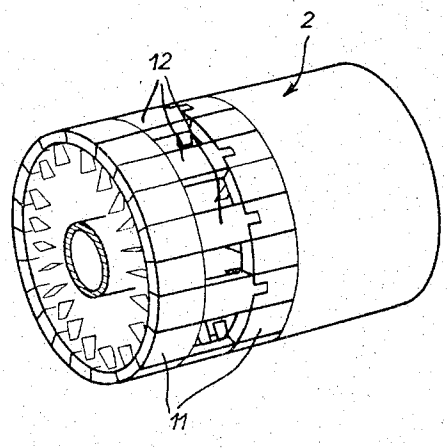

Fig. 5

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United States Patent Office

3,363,831

Patented Jan. 16, 1968

3,363,831

AXIAL-FLOW COMPRESSOR WITH TWO CONTRA-ROTATING ROTORS

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Filed June 23, 1966, Ser. No. 559,771
Claims priority, application France, June 24, 1965, 22,159
8 Claims. (Cl. 230—123)

ABSTRACT OF THE DISCLOSURE

A multistage axial-flow compressor having two contra-rotating rotors with several interdigitated blade-rings alternately driven from the outer periphery thereof in one rotor and from the inner periphery thereof in the other rotor, the outwardly driven blade-rings being operatively inter-connected by an assembly which comprises a circumferential succession of contiguous blade-root platform elements along the outer periphery of each of said latter blade-rings to form a continuous circular casing section with inwardly projecting blades integral therewith, and a plurality of spacers extending from one of said casing sections to the next adjacent one to bridge the gap therebetween, said spacers engaging the corresponding casing sections to transmit thereto tangential forces for rotating the same.

The present invention relates to an axial flow compressor of the type having two coaxial rotors turning in opposite directions and having imbricate or interdigitated blade-rings which belong alternately to one rotor and the other. It derives from the following general preliminary considerations regarding the respective characteristics of axial flow compressors.

It is known that a conventional axial flow compressor composed of a rotor and a stator is capable of a pressure ratio of 1.25. This result is achieved with a transonic rotor the peripheral velocity of which is 330 metres per second and the maximum relative Mach number of which, for the first revolving rotor section or wheel, is 1.25. With a transonic compressor the peripheral velocity of which amounts to 400 metres per second, the maximum pressure ratio that can be hoped for from such a compressor is of the order of 1.45 with a maximum relative Mach number of 1.40 and an efficiency of from 0.80 to 0.85.

An axial flow compressor of the same type in which the relative maximum Mach number was to be supersonic, with a peripheral velocity of 450 metres per second, would be capable of reaching a pressure ratio of 1.60 with an efficiency lower than that of the above compressor and not exceeding 0.78.

A compressor with two coaxial rotors having imbricate blade-rings that contra-rotate, in which compressor the normal rotor has a peripheral velocity of 400 metres per second and the second rotor a velocity of 250 metres per second as it rotates in the opposite direction to the first, would be capable of a pressure ratio of 1.80 with respective relative Mach numbers of 1.35 and 1.15 and an efficiency probably slightly lower than that of a coaxial compressor of conventional type with two stages having the same pressure ratio.

It can be calculated that the gain achieved is on average of the order of 30 to 40%.

In these conditions, it can be estimated that the design of a turbo-machine with a given pressure ratio, according to this technique, leads to a reduction in the number of stages in the compressor of from 30 to 40%, in relation to an engine the compressor of which is designed according to the conventional technique as regards both rotor and stator.

Now, such a solution, though it has been known for a long time, has not been practiced for two basic reasons arising from difficulties peculiar to the aerodynamics of transonic or supersonic cascades and to rotor technology. However, recent progress in the aerodynamics of transonic axial flow compressors and in those of supersonic cascades give reason to believe that an axial flow compressor with two imbricated blade-ring rotors that contra-rotate could operate safely with high efficiency at pressure ratios which are considerable in each stage.

The technology of modern axial flow compressor rotors has evolved either dependent on the bringing into existence of new materials (steels with superior mechanical characteristics, titanium alloys) or on the creation of new manufacturing processes, such as welding with electron streams, high-precision forging of blades, precision-casting of alloys having superior mechanical characteristics, etc. At the present time, this evolution makes possible embodiments that some years ago would have appeared utopian.

Thus, a knowledge of materials possessing high mechanical strength and new techniques of manufacture have made it possible, by calling on new technological solutions, to obtain reasonable peripheral velocities without which an arrangement of imbricated blade-ring, contra-rotating rotors would have afforded no progress in comparison with the conventional solution.

The present invention supplies a solution to technological problems posed by the axial flow compressor having two imbricated blade-ring, contra-rotating rotors.

In conformity with the present invention, the blade-rings of one of the rotors are driven in rotation through their outer periphery by degrees from one ring to the next successive one, by means of an assembly that makes possible their imbrication with the blade-rings of the other rotor, which can be driven in rotation through their inner periphery in the usual way, said assembly being constituted, on the one hand, by contiguous blade roots in circumferential succession along the outer periphery so as to constitute a circular collar or casing section for each of the stages of the rotor in question and, on the other hand, by a series of suitably profiled spacers adapted to transmit tangential driving forces the torque of which is applied to one end of the rotor concerned.

In the drawings:

FIG. 1 is a diagrammatic axial section through a turbo-fan engine including a compressor in accordance with the present invention,

FIGS. 2 and 3 are perspective views partly in section of a portion of the compressor casing, and

FIGS. 4 and 5 are diagrammatic views in perspective of two variants embodying the invention.

The turbo-fan engine shown in FIG. 1 comprises two contra-rotating coaxial rotors **1** and **2** having imbricated or interdigitated blade-rings. The rotor marked **1**, which may, if so required, include a blower **3** for dilution air, is driven by the low-pressure turbine **4** through the agency

of a shaft 5. The other rotor 2, in view of its coaxial relationship with the first rotor 1 and the interdigitated relationship of the blade-rings of the two rotors, must resort to outside drive for its blade-rings, originating from a shaft 7, the necessary power being produced by the high-pressure turbine 6. This latter may equally well drive a stage 8 complementary to the blower 3 for dilution air, which forces a flow into a by-pass duct 9. The motive gases are supplied to the turbine group 6 and 4 by way of a combustion chamber 10.

In such a rotor 2, the centrifugal stresses on the blades can be taken up:

by an outer collar, the blades then functioning in compression and the top peripheral velocities attainable being 180 metres per second for steel and 250 metres per second for titanium;

or by discs, as in the conventional solution, the blades then operating in traction, the peripheral velocities attainable being then 250 metres per second for steel and 350 metres per second for titanium;

or, thirdly, by a combination of the two methods, so long as the system remains isostatic.

In the first case, with the blades fixed in the outer collar, clearly these blades are not only supported but also rotated with the collar. On the contrary, in the arrangement involving discs—the one most advantageous from the point of view of peripheral velocities, the simplest one and the lightest—the blades must be driven from the outside separately from their support discs, by degrees from one ring to the next successive one, by straddling the interdigitated blade-rings of the first rotor 1.

The assembly to which the present invention relates, includes (see FIGS. 2 and 3) compressor blades having contiguous blade roots 11 forming a circumferential succession of platform elements along the outer periphery of each blade-ring of rotor 2 and they constitute a circular collar or casing section. The connection of one such blade-ring to the next for the transmission of tangential driving forces, the torque of which is applied either to the front end or to the rear end of the rotor, is ensured by a series of spacers 12 having a special profile.

On one side, the spacer 12 is welded at 13 to the root elements 11 of a blade ring belonging to one stage and on the other side it is designed to engage the root elements 11 of the blade-ring of the following stage. The spacer 12 is engaged by forming suitable crenellations 14 in the spacer and in the casing section of the corresponding blade-ring. The crenellations are arranged in such a way that a male element on the spacer corresponds to a female element on the casing section, and vice versa.

The arrangement has a degree of play, so that it remains always isostatic whatever the operating conditions (temperatures and rotation velocities), in spite of lengthening of the blades (thermal or mechanical lengthening) and/or the axial expansion of the rotor.

The play under discussion is that existing at the junction of the spacers and of the casing section formed by the blade roots, both circumferentially (between the sides of crenellations) and radially (due to radial offset of the crenellations along the longitudinal axis of the engine). This offset allows degrees of play to be preserved while fluid leaks are minimized.

In order to reduce as far as possible the additional centrifugal stresses on the blades resulting from the presence of the spacers, the latter may be cast in a high-strength alloy. Their cross-section will possess maximum inertial moment transversely, by a judicious arrangement of cavities formed in the casting process, so that the spacers obtain maximum relief.

Finally, the spacers may, if so required, constitute a continuous sleeve, so that the rotor constitutes a homogeneous drum (FIG. 4) instead of a squirrel cage (FIG. 5).

It should be noted that each of the spacers, which form the various bridge connections between the successive rotor stages, is welded to the casing section, formed by the blade roots of the following or preceding stage, by a single circular welding bead. Not every root, therefore, must have a spacer corresponding to it, and the joins of the spacers inter se and of the blade roots inter se are not necessarily opposite each other; indeed, the number of blades may vary from one blade-ring to another.

In the drum-type rotor, the spacers form a continuous sleeve (even though the spacers are not welded inter se on their lateral surfaces). The spacers are however, circumferentially spaced in a squirrel cage rotor. Leakages of fluid are greater than in the preceding case but they are still tolerable.

FIGS. 4 and 5 represent cylindrical rotors with spacers and blade roots that cooperate with each other.

However, the fluid path may be conical, as in FIG. 1. The joins may also be staggered, as indicated above. In this case the crenellations will not be located identically on every spacer or blade root.

What I claim is:

1. A multistage axial-flow compressor having two contra-rotatable rotors with several interdigitated blade-rings alternately driven in rotation from the inner periphery with regard to the blade-rings of one rotor and from the outer periphery with regard to the blade-rings of the other rotor, wherein the improvement comprises a mechanical assembly for operatively interconnecting said latter-mentioned blade-rings through the outer periphery thereof, said assembly comprising a circumferential succession of contiguous blade-root platform elements which form together a continuous circular casing section with inwardly projecting blades integral therewith, and a plurality of spacers extending axially between successive casing sections and engaging the same for the transmission thereto of tangential driving forces, whereby said latter-mentioned rotor is bodily rotated.

2. Compressor as claimed in claim 1, further comprising an annular disc carrying along its outer periphery each said latter-mentioned blade-rings through the inner periphery thereof, and bearing means for supporting said annular discs for free rotation along the inner periphery thereof.

3. Compressor as claimed in claim 2, further comprising a driving shaft for the former-mentioned blade-rings, threaded through said annular discs, said bearing means being interposed between said shaft and the inner periphery of said discs.

4. Compressor as claimed in claim 1, wherein each spacer is, at one axial end thereof, integral with the adjacent side of a casing section of one blade-ring of said latter-mentioned rotor and bears, at the other axial end thereof, against the adjacent side of the casing section of the next successive blade-ring of said latter-mentioned rotor.

5. Compressor as claimed in claim 4, wherein said other axial end of each spacer and said adjacent side of the casing section of said next successive blade-ring are mutually shaped for reciprocal clutch engagement.

6. Compressor as claimed in claim 5, wherein said clutch engagement has a degree of play both circumferentially and radially.

7. Compressor as claimed in claim 1, wherein the spacers between successive casing sections form continuous sleeves extending all around said latter-mentioned rotor, whereby said assembly assumes the shape of a continuous barrel.

8. Compressor as claimed in claim 1, wherein the spacers between successive casing sections are circumferentially spaced from each other, leaving circumferentially spaced gaps in between, whereby said assembly assumes the shape of a squirrel cage.

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