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(54) **AXIAL FLOW TURBO COMPRESSOR**

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(57) **ABSTRACT**

An axial flow compressor including one or more axially spaced stator sections **(14)** each having a circumferential array of guide vanes **(15)**, and one or more rotor section (12, 16) each having a circumferential array of rotor blades **(13, 17;** A, B), and between successive rotor blades (A, B) there are formed flow paths and between the stator and rotor sections there are provided axial gaps **(22, 23, 24)** of a diverging configuration so as to form in each flow path a diffusion region (C) which extends from a narrowest cross section (a_2) of the flow path located at a certain distance upstream of the leading edge of a second rotor blade (B) to a wider cross section (a_3) located approximately at the leading edge of the second rotor blade (B), wherein each flow path has a transition region (D) of a substantially non-increasing cross sectional area extending from the wider cross section (a_3) to the trailing edges of the rotor blades (A, B).

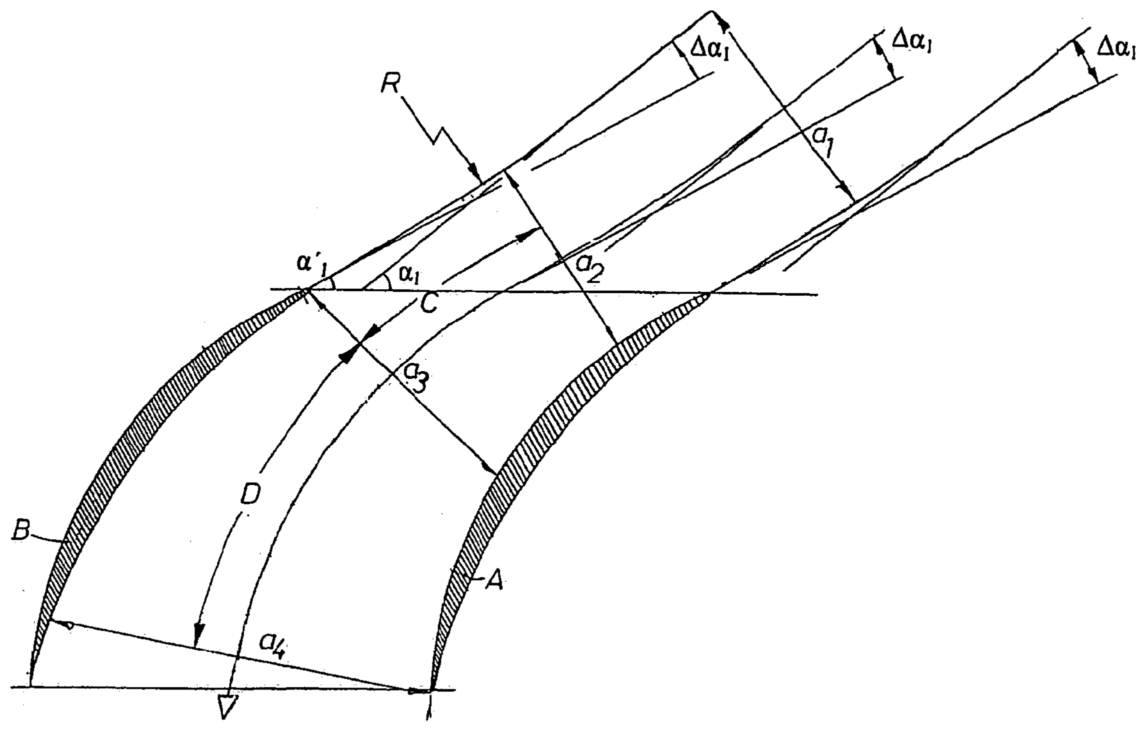
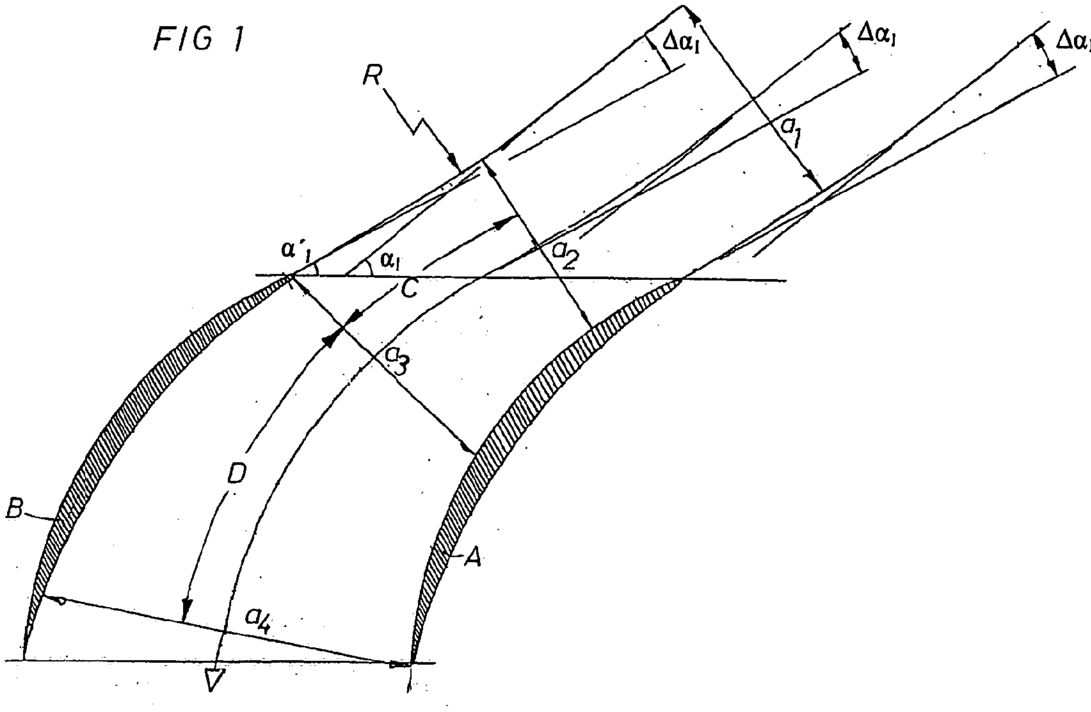
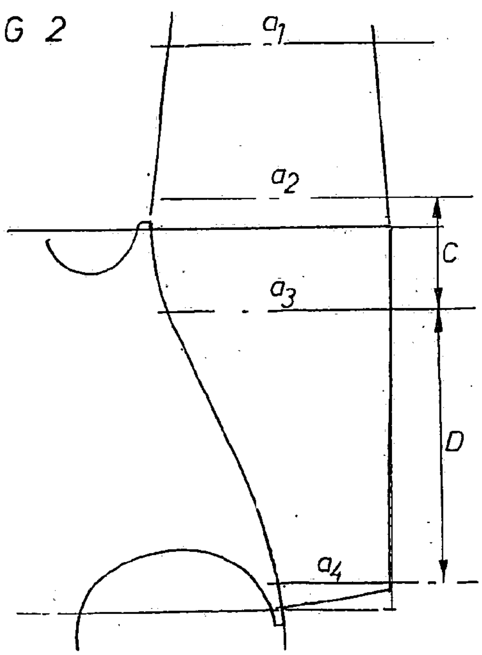


FIG 1



R
$\Delta\alpha_1$
$\Delta\alpha_1$
$\Delta\alpha_1$
a_1
a_2
α'_1
α_1
C
a_3
D
B
A
a_4

FIG 2



a_1
a_2
C
a_3
D
a_4

FIG 3

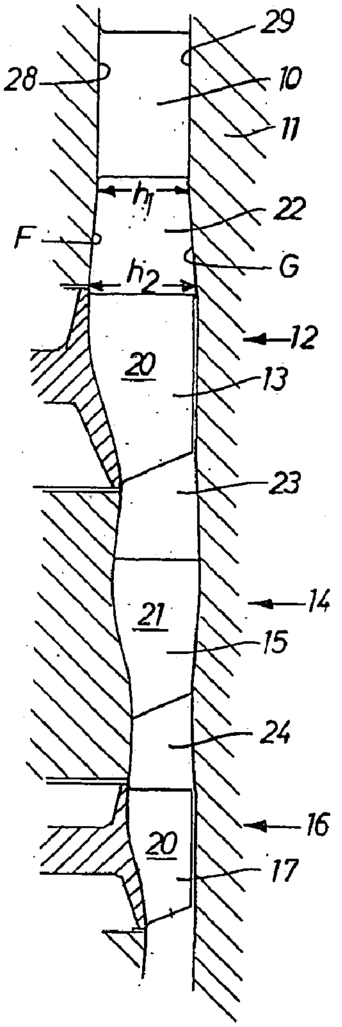
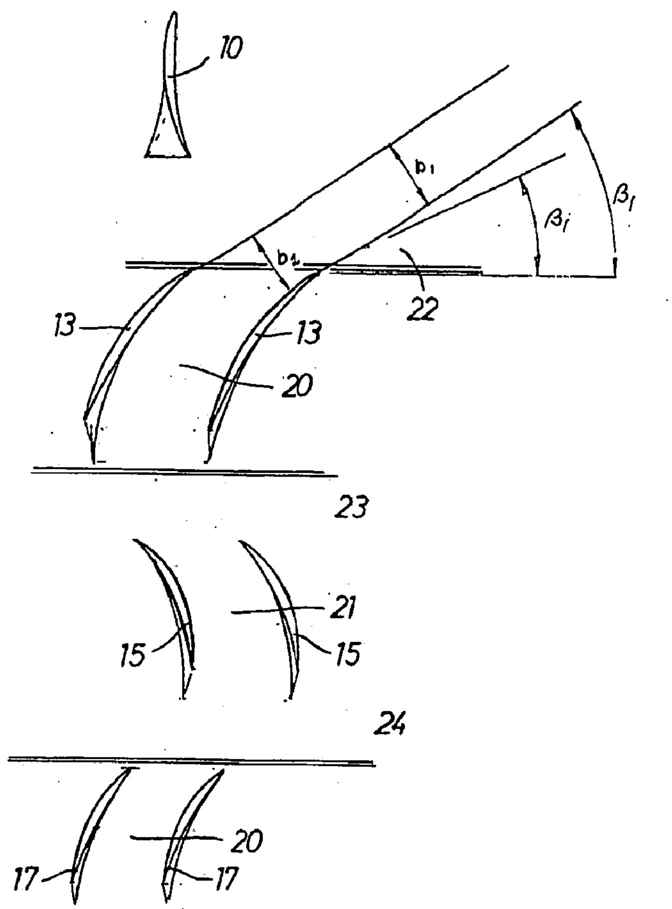

29
28
10
11
h_1
22
F
G
h_2
12
20
13
23
14
21
15
24
16
20
17

FIG 4


10
b1
β_l
β_i
b2
22
13
13
20
23
21
15
15
24
20
17
17

AXIAL FLOW TURBO COMPRESSOR

[0001] The invention relates to a turbo compressor of the axial flow type comprising a stator with at least one axial section including a circumferential array of flow directing guide vanes and a rotor with at least one axial section including a circumferential array of rotor blades, wherein between the guide vanes and the rotor blades and between an inner peripheral wall and an outer peripheral wall there are formed parallel flow paths, and between successive rotor blades there are formed rotor flow passages through which the flow paths extend.

[0002] In prior art compressors of the above type there is a problem of obtaining an increased pressure ratio across each compressor stage and/or an increased efficiency. A factor which is limiting and crucial for these objectives is the mean velocity of the air flow through the compressor. It is a well known fact that an increased air flow velocity would give a higher pressure ratio across each compressor stage and/or an increased efficiency. In prior art compressors, however, the flow velocity is kept well below the sonic velocity, i.e. a Mach number below 1.0, usually around 0.7, because at super sonic velocity there arise shock waves in the air flow which are difficult to avoid and which are detrimental to the pressure ratio and the compressor efficiency. By keeping the Mach number around 0.7 there is ensured that the Mach number 1.0 will not be reached and that no shock waves will arise.

[0003] The reason for using such a large "safety" margin of 0.7-1.0 in Mach number is that in prior art compressors the air flow velocity normally increases locally at the downstream ends of the flow paths of the stator or rotor sections. The reason for such velocity increase is that when departing from a flow path between two guide vanes or two drive blades the air flow is subjected to a tangential contraction due to a change in flow direction. Such an increase of the flow velocity might bring the air flow velocity up to a Mach number around 1.0, and undesired shock waves might arise in the air flow. In order to make sure that sonic velocity is not reached at any location in the compressor, the air flow velocity is kept down to the "safe" Mach number 0.7.

[0004] There are transonic compressors working at velocities exceeding Mach number 1.0 and by which special arrangements have been made to avoid the negative influence of shock waves. However, that type of compressor would also benefit from lower air flow losses in the stator and rotor flow paths in accordance with the invention.

[0005] The main object of the invention is to accomplish a compressor of the above type working at subsonic air flow velocities and where the air flow passages through the compressor are improved in such a way that the mean air flow velocity through the stator and rotor sections may be increased considerably without risking the Mach number reaching the 1.0 level.

[0006] Further objects and advantages of the invention will appear from the following specification and claims.

[0007] A preferred embodiment of the invention is hereinafter described in detail with reference to the accompanying drawing.

[0008] On the drawing

[0009] **FIG. 1** shows the geometry of the flow path through a rotor blade passage.

[0010] **FIG. 2** shows a side view of the rotor blade passage of **FIG. 1**.

[0011] **FIG. 3** shows a fractional longitudinal section through a compressor according to the invention.

[0012] **FIG. 4** shows a spread-out view of the rotor blades and stator guide vanes of the compressor illustrated in **FIG. 1**.

[0013] In **FIG. 1** there is illustrated a flow path relative to the rotor which extends through a passage between two successive rotor blades A and B. Before entering the passage between the rotor blades A,B, the medium flow is deflected in a direction opposite the movement direction ω of the rotor blades by an angle $\Delta\alpha_1$, which is the difference between the original flow path angle α_1 and the new flow path angle α_1. This deflection of the flow causes a sidewise contraction of the flow path and follows a curvature which may be theoretically calculated via a well recognised method, see for instance: Eckert/Schnell "Axial-und Radialkompressoren", 2:nd edition, page 264, or "Dubbel Taschenbuch für den Maschinenbau", 1974, page 334. The curvature has a shape which is close to a circle line with a radius R.

[0014] When reaching a section a_2 at a certain distance upstream of the leading edge of the second rotor blade B or slightly downstream of the leading edge of the first rotor blade A the flow path passes through a diffusor region C which extends in the flow direction to a section a_3 approximately at the leading edge of the second rotor blade B. Accordingly, the diffusor region C has an entrance section a_2 and an exit section a_3, wherein the entrance section a_2 has a cross sectional area which is smaller than that of the exit section a_3. The entrance section a_2 of the diffusor region C is also the narrowest cross section of the entire flow path between a_1 and a_4.

[0015] Downstream of the diffusor region C, the flow path extends through a transition region D which has a substantially non-increasing or slightly decreasing cross sectional area all the way from section a_3 to an exit section a_4. To compensate for a downstream increasing distance between the rotor blades A and B the radial extent of the rotor blades, i.e. the radial distance between the inner peripheral wall **28** and the outer peripheral wall **29**, has to be reduced so as to keep the cross sectional area substantially constant throughout the transition region D. See **FIG. 2**. In some cases it might be advantageous to have a slight acceleration of the flow through the transition region D.

[0016] Upstream of the diffusor region C, the flow path has a substantially constant cross sectional area, from an initial section a_1 to the diffusor region entrance section a_2 so as to generate a non-increasing flow velocity. As illustrated in **FIG. 2**, this is accomplished by forming the inner and/or the outer peripheral walls **28**,**29** of the rotor and the stator, respectively, with diverging surfaces F and G. These diverging surfaces F, G compensate for the sidewise contraction of the flow path, as described above, and serves to keep down the Mach-number of the flow velocity and prevent shock waves to occur in the medium flow.

[0017] By locating the diffusor region C of each flow path upstream of the flow deflecting transition region D between two successive rotor blades A, B there is accomplished a reduction in flow velocity and, hence, a reduction of the flow losses during the flow path deflection between the rotor blades A, B. This means an improved efficiency of the compressor.

[**0018**] In order to ensure a good efficiency of the compressor the flow velocity shall be equally high over the entire radial extent of each rotor blade. This is accomplished by employing a guide vane configuration in the initial compressor stage such that each guide vane **10** has a different flow deflection angle at its top end compared to its bottom end. See **FIG. 4**. Thereby, there is obtained optimum flow directions for generating an equal flow velocity at all radial locations on each rotor blade in the initial compressor stage.

[**0019**] In **FIGS. 3 and 4**, there is illustrated a preferred embodiment of the invention including the flow path characteristics illustrated in **FIG. 1**.

[**0020**] In **FIG. 3** there is shown a sectional view of an inlet nozzle for the initial stage of the compressor including guide vanes **10** rigidly mounted in a housing **11**. Downstream of the nozzle **10** there is a rotor section **12** with a rotor blade **13** followed by a stator section **14** having a guide vane **15** secured to the housing **11**, and another rotor section **16** with a rotor blade **17**. Rotor flow paths **20** extend between two adjacent rotor blades **13,17**, and stator flow paths **21** are formed between two adjacent guide vanes **15**. The flow paths **20**, **21** are also defined by an inner peripheral surface **28** and an outer peripheral surface **29**.

[**0021**] Between the stator sections and the rotor sections there are provided axial gaps which form annular air flow passages **22**, **23** and **24**.

[**0022**] The main character of the air flow passage through the compressor is successively converging from the inlet nozzle end toward the outlet end. As illustrated in **FIG. 3**, the cross sectional area of the air passage decreases stepwise. In the air flow paths **20** between the rotor blades **13** as well as the flow paths between the guide vanes **15** the radial extent of the flow passage decreases, whereas in the flow passages **22**, **23** and **24** located between the stator sections **14** and the rotor sections **12** the radial extent of the flow passage increases.

[**0023**] A characteristic feature of the invention is the provision of the axial gaps between the stator and rotor sections forming the flow passages **22**, **23** and **24**. The reason for introducing these axially extended and radially diverging passages **22**, **23** and **24** is to accomplish a velocity reducing diffusor region with the purpose to reduce flow losses and increase the compressor efficiency.

[**0024**] As illustrated in **FIG. 4** an air flow approaching the rotor flow path **20** between two rotor blades **13,17** has a converging shape, because depending on a difference in direction between the incoming air flow and the direction of the rotor blades **13,17**, the air flow has to change direction. As illustrated in **FIG. 4**, the direction of the incoming air flow path forms an angle to a radial plane and is denoted β_1. This angle is larger than the angle of the rotor blades **13,17**, which is denoted β_1. Due to this change in flow direction, the air flow path is subjected to a tangential contraction, which causes an increased flow velocity. This is illustrated by the numerals b_1 and b_2, where b_2 illustrates a narrower flow path cross section than the incoming flow in section b_1. The acceleration of the air flow is disadvantageous since it results in an increased frictional losses in the flow paths.

[**0025**] This undesirable acceleration of the air flow is omitted by increasing the available cross sectional area in the flow passage, i.e. by the introduction of the intermediate and radially diverging flow passages **22**, **23** and **24**. By increasing the radial extent of these passages by at least 10% there is obtained an improved compressor efficiency. For obtaining a substantial increase in the compressor efficiency the increase in the radial extent of the flow passages **22**, **23** and **24** should be at least **20**%. In the illustrated example, the radial extent of the passages increases from h_1 at the entrance to h_2 at the exit end.

[**0026**] For obtaining a favourable shape of the air flow path through the compressor, the increase in radial extent of the intermediate passages **22**, **23** and **24** has to be accomplished over a certain passage length. Therefore, the passages **22**, **23** and **24** should have an axial length exceeding 30% of the rotor blade and guide vane length, respectively. Depending on the radial extent of the blades and vanes the passage length could be 50% or more of the length of the blades and vanes, respectively.

1. Axial flow turbine compressor, comprising a stator with at least one axial section including a circumferential array of flow directing guide vanes (**13,17**), a rotor with at least one axial section including a circumferential array of rotor blades (**15**; A, B), an inner peripheral wall (**28**), and an outer peripheral wall (**29**), wherein a flow path is formed between every two successive rotor blades (A,B) in the direction of rotor rotation (ω) and between said inner and outer peripheral walls (**28,29**), characterized in that each flow path comprises

a narrowest cross sectional area at a first cross section (a_2) located at a certain distance upstream of the leading edge of a second rotor blade (B) in the direction of rotor rotation (ω),

a diffusion region (C) having a successively increasing cross sectional area in the flow direction and extending from said first cross section (a_2) to a second cross section (a_3) located approximately at the leading edge of said second rotor blade (B), and

a transition region (D) extending in the flow direction from said second cross section (a_3) to the trailing edge of said second rotor blade (B), said transition region (D) has a substantially non-increasing cross sectional area in the flow direction throughout its length.

2. Turbine compressor according to claim 1, wherein the transverse distance between said first and second rotor blades (A,B) increases throughout said transition region (D), whereas the radial distance between said inner peripheral wall (**28**) and said outer peripheral wall (**29**) decreases such that the cross sectional area of the flow path does not increase throughout said transition region (D).

3. Turbine compressor according to claim 1 or **2**, wherein the flow through said diffusion region (C) is substantially laminar.

4. Turbine compressor according to anyone of claims **1-3**, wherein each flow path upstream of said diffusion region (C) has a substantially constant cross sectional area.

5. Axial flow type turbine compressor, comprising a stator with at least one circumferential row of flow directing guide vanes. (**13**,**17**), and a rotor with at least one circumferential row of rotor blades (**15**; A,B), wherein between said guide vanes (**13**,**17**) and said rotor blades (**15**; A,B) there are formed a number of parallel flow paths, and between successive rotor blades (A,B) there are formed rotor flow passages through which said flow paths extend, characterized in that each flow path has a diffusion region (C) extending from a location approximately at the leading edge of a first one (A) of said successive rotor blades (A,B) to a location approximately at the leading edge of a second rotor blade (B) in the rotation direction (ω) of said rotor, wherein each one of said rotor flow passages downstream of said diffusion region (C) has a substantially non-increasing cross sectional area throughout its length.

6. Turbine compressor according to claim 5, wherein said flow paths as well as said rotor flow passages are partly defined by an outer peripheral wall (**29**) and an inner peripheral wall (**28**).

7. Turbine compressor according to claim 5 or **6**, wherein each one of said rotor flow passages is defined by a first rotor blade (A), and a second rotor blade (B) succeeding said first rotor blade (A) in the direction of rotor rotation (ω), said diffusion region (D) extends in the flow direction from a location approximately at the leading edge of said first rotor blade (A) to a location approximately at the leading edge of said second rotor blade (B), and each one of said flow paths has a narrowest cross sectional area (a_2) at the upstream end of said diffusion region (C).

8. Turbine compressor according to claim 1, wherein said guide vanes (**13**,**17**) are arranged in two or more axially spaced stator sections, and said rotor blades (**15**) are arranged in two or more axially spaced rotor sections,

said rotor sections and said stator sections are arranged with an axial gap (**22**,**23**,**24**) between them,

said axial gap (**22**,**23**,**24**) having an axial width of at least 30% of the chord length of the preceding guide vane (**13**,**17**) or rotor blade (**15**), and

said axial gap (**22**,**23**,**24**) forms a flow passage region with a,radially diverging shape in the axial direction.

9. Turbine compressor according to claim 8, wherein said flow passage region has a radial extent (h_2) at its downstream end that is at least 10% larger than the radial extent (h_1) at its upstream end.

10. Turbine compressor according to claim 8, wherein said flow passage region has an axial width of at least 50% of the chord length of the preceding guide vane (**13**,**17**) or rotor blade (**15**).

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