March 11, 1941. G. JENDRASSIK 2,234,733

COMPRESSOR OR PUMP OF THE ROTARY BLADES TYPE

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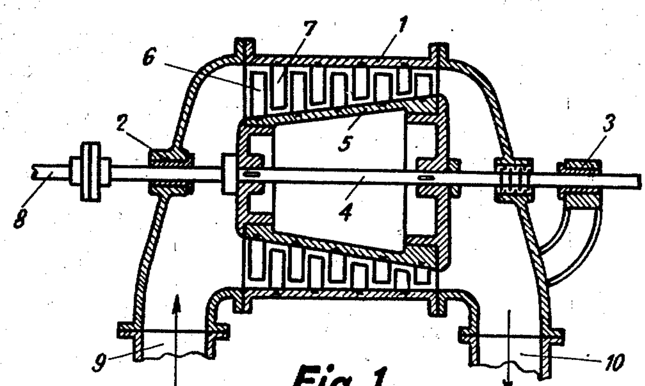

Fig. 1.

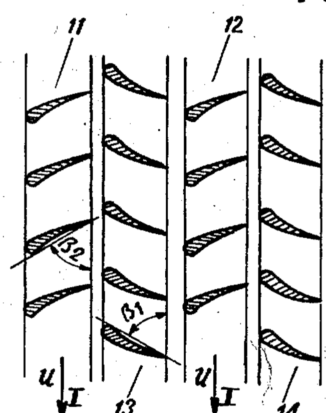

Fig. 2.

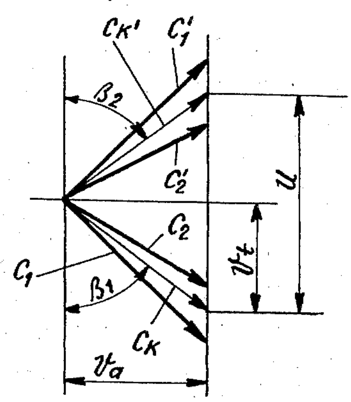

Fig. 3.

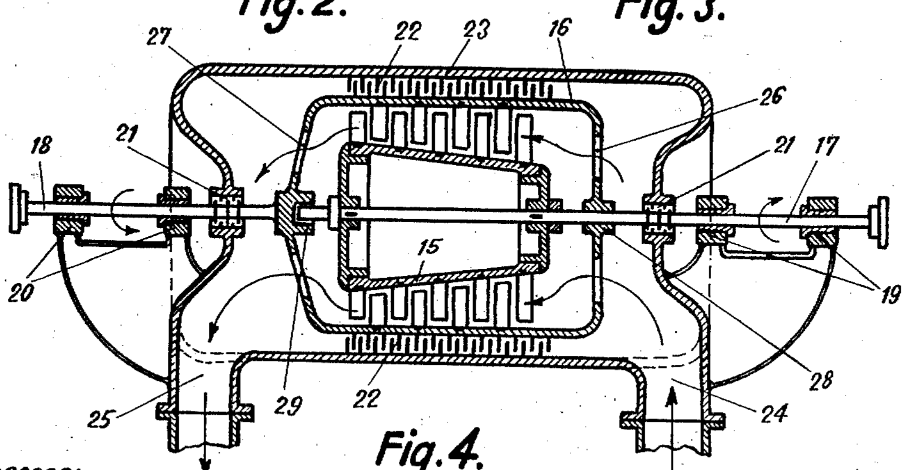

Fig. 4.

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COMPRESSOR OR PUMP OF THE ROTARY BLADES TYPE

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6 Claims. (Cl. 230—122)

The invention relates to a compressor or pump of the rotary blades type in which the mean diameter of any stationary or rotating blade ring is at least approximately equal to the average of the mean diameters of the two blade rings adjacent to it. This type includes, for instance, axial throughflow compressors and pumps, or compressors and pumps of a kind in which the flow of the working medium takes place along a surface of rotation having a straight or curved generatrix, as for instance along a cone surface, and the blades of which have air foil cross-sections or are of thin sheet-like cross-section (vane-shaped). In the case of compressors and pumps of this type as known up to now the drawback has presented itself that the friction of the working medium taking place on the surface of rotation guiding the flow, as on the compressor casing and on the wall of the rotor, as well as on the blades, has exerted a disadvantageous influence on the distribution of velocities of the working medium, and for this reason the efficiency of these machines and the pressure which it was possible to produce by their means have for many applications not been sufficient.

The layer of medium alongside the walls constituting the boundaries of the flow or on those parts of the blades which are adjacent to these boundary walls, which layer of medium is braked by the friction or by having become detached from the suction side of the blades, i. e. the so-called "tired" boundary layer does not possess the same relative velocity relatively to the blade ring as possessed by the portions of medium contained in the sound core of the flow and therefore the blades are unable to produce a suitable rise of pressure in these places; accordingly the greater the success achieved in eliminating the tired boundary layers or at least reducing their dimensions, the more perfectly the compressor or pump of the rotary blades type will be able to operate. The boundary layer is, on the one hand, liable to rub against the blades and against the walls, whereby its speed is influenced in a disadvantageous manner, whilst on the other hand it is in a frictional relation with the sound core of the flow likewise, by which latter friction, however, the velocity conditions of the boundary layer are influenced in an advantageous manner. In the compressor or pump of the rotary blades type according to the invention this drawback is eliminated by constructing the blading in such a manner, that the impulse convection (friction) between the sound core of the flow and the boundary layer diminished in its relative speed relatively to the blade ring should be substantially stronger than in the case of the types of apparatus known up to now.

In order to enable the fundamental idea of the invention to be more readily understood, let us suppose that a mass portion m of working medium possessing the mass m possesses a velocity component v_t in the peripheral direction, the said component suffering owing to friction either against the stationary or against the moving boundary wall or owing to friction against the blade rings, a variation of the infinitesimal figure of dv_t. The centrifugal force acting on this particle of medium is

$$P=\frac{mv^2_t}{r}$$

in which formula r means the distance from the axis of rotation. Owing to the said infinitesimal alteration of the peripheral component of the velocity this centrifugal force suffers an infinitesimal variation amounting to

$$dP=\frac{2m}{r}v_t dv_t$$

With a given variation of velocity dv_t, this variation of the centrifugal force is all the greater, the greater the peripheral velocity component of the particle of medium. If in order to enable the conditions to be imagined more clearly it is supposed that the working medium, for instance in the case of a compressor of axial throughflow, possesses an average velocity of rotation of a direction identical with that of the peripheral velocity of the rotor, but of smaller magnitude than this velocity, that part of the working medium which rubs against the wall of the rotor will become accelerated and the surplus of centrifugal force acting on it in consequence hereof will be all the greater, the greater the rotation of the working medium has been. Similarly, if the working medium rubs against the stationary wall (against the internal surface of the casing of the machine), its velocity will become diminished and the diminution of the centrifugal force acting on it will be all the greater, the greater the average rotation of the medium has been. The force increment dP_c acting on the particle rubbing against the rotor will drive the rubbing particles outwards with a substantial force, whilst the particles of medium rubbing against the stationary casing will be impelled strongly towards the axis of rotation by the diminution of the centrifugal force as compared to the ambient. Both effects will give rise to a high amount of convection, the condition for this being accordingly to keep the working medium in a substantial average rotation of not negligible extent relatively to the peripheral velocity of the rotor, the direction of such rotation being identical with the direction of the peripheral velocity of the rotor, but of smaller magnitude than the latter. The working medium will—as is the case with the compressors or pumps of the rotary blades type known up to now—hardly possess any speed component in the peripheral direction and thus the variations of such component resulting in consequence of the friction caused by the centrifugal force will in the case of small variations of velocity be very small only, and in order that they should reach any appreciable figure, a variation of velocity of very great extent is necessary. The creation of increased exchange of impulses will possess particularly great importance in case the machines according to the invention are of the multi-stage type, or, if the pressure head to be produced is high relatively to the velocity head corresponding to the peripheral velocity of the rotor. In the case of fans, where, for instance, only one or two blade rings are employed, the importance of the friction of the boundary layer and, accordingly, also the importance of increasing the amount of convection, are substantially smaller.

In order to enable the invention to be more readily understood, Fig. 1 of the accompanying drawing represents in a diagrammatical longitudinal section an embodiment shown by way of example of such a compressor or pump, Fig. 2 shows the picture, developed into a plane, of a section taken through the blading, Fig. 3 shows the corresponding velocity triangles of the stationary and of the moving blades, whilst Fig. 4 shows another embodiment of the invention having rotating guide blades.

In Fig. 1, the rotor **5**, in casing **1**, is fixed on shaft **4** journalled in bearings **2** and **3**. The rotor carries the rotary blade rings **6**. It is in the compressor casing that the stationary blade rings **7** are accommodated. The shaft end **8** serves for driving the compressor. This apparatus operates in such a manner that the rotor will, if set into rotation in the proper direction, draw in the working medium through the inlet opening **9** and discharge it in compressed condition through the opening **10**. On Fig. 2 the rotating blade rings **11** and **12** move in the direction of the arrow I (in the plane of the section) at the peripheral velocity u, whilst the stationary blade rings **13** and **14**, remain in a position of rest. The base line of the profile of the blades (its tangent line or in the case of blades convex on both sides the tangent of the adaptation circles of the blade profile tips) is at the outlet edge of the moving blades forming the angle β_2 with the peripheral direction, whilst at the outlet ends of the stationary blades the angle formed by this base line and the peripheral direction is β_1. When determining the angles β_1 and β_2, that one of the angles formed at the outlet end of the blade by the base line and the peripheral direction should be taken, between the sides of which the section of the blade does not fall, or, (for instance in the case of blades convex on both sides), that one between the sides of which only the smaller portion of the blades is situated. In this manner the angles are defined unequivocally (they should always be measured on the working face of the blades at the outlet from the blade ring), and in what follows also, it is always the angles measured in this manner which are meant when reference is made to blade angles.

On Fig. 3 c_1 denotes the absolute velocity of the working medium before entrance into the stationary blades, whilst c_2 denotes the absolute velocity of the said working medium after outlet from the stationary blades; the mean absolute velocity of the working medium is c_k. The component of peripheral direction of this last-named velocity, which component is equal to the average velocity of rotation of the medium, is v_t; whilst the component v_a perpendicular thereto is the meridian velocity, which in the case of a compressor of axial throughflow is equal to the axial velocity. The absolute velocities at the same time also mean the relative veocities relatively to the stationary blades. The velocities relatively to the moving blades are obtained by adding the peripheral velocity u of the rotor, in the proper direction, to the velocities referred to above. Thus c'_1 is the relative velocity as between the medium and the rotating blades before the entrance of the medium into the rotating blades, c'_2 is the relative velocity after the medium has left the rotating blades, whilst c'_k is the main relative velocity.

As in practice the base line of the blades is with a good approximation equal to the mean direction of velocities, the angles β_1 and β_2 shown on Fig. 3 are identical with the angles β_1 and β_2 shown on Fig. 2.

As has been explained in what precedes, in order that a strong exchange of impulses should take place between the boundary layer and the core of the flow it is necessary that the average peripheral velocity v_t of the medium should be sufficiently high relatively to the peripheral velocity u. Thus a fairly good result can already be obtained if the average velocity of rotation will at least on one diameter along the length of the blades reach one-third of the peripheral velocity. Naturally this exchange of impulses will be more advantageous if the average peripheral velocity of the medium is higher still than the figure referred to, by way of a limiting value it may even rise to a figure equal to the peripheral velocity of the rotor, naturally with the condition that its direction always remains equal to this last-named direction. The constructional condition for ensuring that it should be possible for the average velocity of rotation of the medium to remain between these two limits, is, as can be deduced also from Fig. 3, that the value of the quotient

$$\frac{\tan \beta_1}{\tan \beta_2}$$

should, at least on one diameter along the blade length, be smaller than 2 and larger than zero, wherein the angles β_1 and β_2 belong to cooperating adjacent stationary and rotating blade rings.

In view of the fact that the magnitude of the meridian component of the velocity also exerts an influence on the attainment of high efficiency, and in view of the fact that the magnitude of this meridian component should on at least one diameter along the length of the blades preferably remain between one-fourth of the peripheral velocity and the total peripheral velocity, it is possible to obtain a compressor of high efficiency in such a manner, if the constructional conditions of this last-named proportion are likewise fulfilled. In case the condition referred to above relating to the average rotation of the medium has also been satisfied, this further condition is fulfilled if the tangent of the angle β_1 between the peripheral direction and the base line of some stationary blade ring is, at least on one diameter along the blade length, greater than ¼ and smaller than 3.

The circumstance that a substantial average velocity of rotation is imparted to the working medium offers substantial advantages from the point of view of the operation of the compressor in case a rotor of high peripheral velocity is concerned. As well known, it is not advisable to allow the relative velocity as between the working medium and the blades to approximate the speed of propagation of sound vibrations in the working medium too closely, as in this case the efficiency of the blades is liable to deteriorate in a substantial extent. In the case of a rotor having a given peripheral velocity, the only way in which the relative velocity can be diminished in a substantial extent consists in giving to the working medium a substantial average velocity of rotation in the direction of rotation of the rotor. This circumstance will, accordingly, on the one hand permit a higher efficiency to be obtained, whilst on the other hand it will render higher peripheral velocities possible with good efficiency. The situation will be particularly favourable from the point of view of this last-named circumstance, but will also be very advantageous from the point of view of impulse exchange, in those cases, when an average rotation of such magnitude is imparted to the working medium as will, at least on one blade diameter along the length of the blades, be equal to one-half the peripheral velocity of the rotor. The constructional condition for this is that, at least on one blade diameter, the angular setting of the stationary and of the rotary blades relatively to the peripheral direction should be mutually equal.

From the point of view of the magnitude of the exchange of impulses it is not immaterial how the average velocity of rotation of the working medium is distributed according to the distance from the axis of rotation. From this point of view it is the most advantageous method to follow the distribution of speed of the so-called potential eddy, in which the velocity of rotation stands in inverse ratio to the distance from the axis of rotation. In the case of such a rotation each particle of the working medium, being in an indifferent position of equilibrium, can be removed to any point of the radius with the smallest virtual force action, and therefore the variation of the centrifugal force acting on any particle will immediately and in the highest degree start convection. The constructional condition for ensuring that the average rotation should take place according to this law is that the ratio of the tangent of the angles β_1 and β_2 should at least approximately follow the law

$$\frac{\tan \beta_1}{\tan \beta_2} = qr^2 - 1$$

in which formula q is a suitably chosen constant, whilst r is the distance measured from the axis of rotation.

In the case of pumps, the rotation of the working medium at a substantial velocity and the diminution by these means of the relative velocity between the working medium and the blades is also advantageous from the point of view of the diminution of the danger of cavitation.

The intensification, in the manner described, of the exchange of impulses also exercises an advantageous effect from the point of view of diminishing the gap loss, seeing that the layer which has suffered a gap loss will become quickly mixed with intact flow and it will not be possible, owing to the gap loss either, for a tired boundary layer, by which the operation of the compressor would be influenced disadvantageously, to develop.

The forms of construction described in the specification are only examples for illustrating the invention, and the invention can be carried into effect in very many kinds of constructional embodiments. From the point of view of interpreting the range of protection of the invention in this manner, it is worth mentioning, for instance, that by a suitable generalisation of the wording of the invention, the range of the invention will also cover compressors constructed in such a manner that, in addition to the rotor, the so-called "stator" (compressor casing) is also made rotatable in a sense of rotation opposite to the sense of rotation of the rotor.

Such an arrangement is shown on Fig. 4 according to which the rotor 15 and the "stator" 16 made rotatable relatively to each other,—the said "stator" being in this case a second external rotor rotating in the opposite sense—are by means of the journalling arrangements 28 and 29 journalled in each other also in such a manner as to ensure that their geometrical axis of rotation should be common. In addition hereto the inner rotor 15 is journalled by means of its shaft 17 in the bearings 19, whilst the external rotor 16 is journalled by means of its shaft 18 in the bearings 20, and the whole is surrounded from the outside by the fixed casing 23 by which the brackets of the bearings 19 and 20 are also supported. This casing 23 is packed relatively to the shafts 17 and 18 by means of the packings 21 which are preferably of the labyrinth type. Packing, preferably labyrinth packing likewise, is required in addition hereto also between the mutually opposite cylindrical surfaces of the external rotor 16 and of the casing 23, in order to prevent that the working fluid entering through the inlet opening 24 of the casing 23, which working fluid has been brought to higher pressure in the compressor, should flow back from the high-pressure space into the low-pressure space of the casing, for which purpose, according to the illustration, the labyrinth packing 22 is employed. The low-pressure working fluid enters the low-pressure space of the casing 23 through the said opening 24, and passes from this space through the opening 26 on the external rotor 16 into the working space proper of the compressor, and streaming between the blades here, leaves the working space of the compressor through the openings 27 of the external rotor, following which it leaves the high-pressure space of the casing 23 through the opening 25. Of course, between the shafts 17 and 18 rotating in mutually opposite senses, some kind of mechanical gear wheel connection, not shown on the drawing, is required for uniting the rotations. If the peripheral velocity in any blade ring of the "stator" (i. e., external rotor) at a certain blade diameter is denoted by u_1, whilst the same peripheral speed in an adjacent blade ring of the rotor proper cooperating with the said stator blade ring is denoted by u_2, u_1 being $\leqq u_2$, it is possible to indicate regarding the ratio of the tangents of the blade angles β_1 and β_2 the following mathematical relation according to the invention:

$$\frac{2}{1+3\frac{u_1}{u_2}} > \frac{\tan \beta_1}{\tan \beta_2} > 0$$

whilst in case the condition established above regarding the meridian component of the flow speed according to which

$$\frac{1}{4} \leq \frac{Va}{u_2} \leq 1$$

is also satisfied, $\tan \beta_1$ should be chosen between the following limits:

$$\frac{1}{4\left(\frac{u_1}{u_2}+1\right)} < \tan \beta_1 < \frac{3}{3\frac{u_1}{u_2}+1}$$

In case of $u_1=0$ (stationary compressor casing) these expressions become converted into the conditions already stated above.

The compressor described is particularly suitable for use in connection with gas turbines, seeing that in the case of gas turbines the high efficiency of the compressor is a very important condition.

I claim:

1. In a multiple stage rotary compressor or pump comprising a rotor and a casing, a plurality of blade rings on the rotor alternating with a plurality of blade rings on the casing, the mean diameter of any blade ring being at least approximately equal to the average of the mean diameters of the adjacent blade rings, the blades of each ring having airfoil profiles and being positioned so that the value of the quotient composed of the tangents of the angles β_1 and β_2 should at least on one blade diameter along the blade length fall within the limits:

$$\frac{2}{1+3\frac{u_1}{u_2}} > \frac{\tan \beta_1}{\tan \beta_2} > 0$$

wherein β_1 and β_2 denote the angles formed at the outlet from the blade ring on the working face between the peripheral direction and the base line of the blade profile (e. g. the line traced tangentially to the working face of the blade profile) of any blade ring on the casing, and of the adjacent blade ring cooperating therewith on the rotor, respectively, u_1 and u_2 denote the corresponding peripheral velocities of these blade rings, the casing and the blades carried thereby being made rotatable in a direction opposite to the direction of rotation of the rotor.

2. In a multiple stage axial flow rotary compressor or pump comprising a rotor and a casing, a plurality of blade rings on the rotor alternating with a plurality of blade rings on the casing, the mean diameter of any blade ring being at least approximately equal to the average of the mean diameters of the adjacent blade rings, the blades of each ring having airfoil profiles and being, at least one one blade diameter along the blade length, positioned so that the value of the quotient composed of the tangents of the angles β_1 and β_2 should fall within the limits:

$$\frac{2}{1+3\frac{u_1}{u_2}} > \frac{\tan \beta_1}{\tan \beta_2} > 0$$

besides the value of $\tan \beta_1$ should fall within the limits:

$$\frac{3}{1+3\frac{u_1}{u_2}} > \tan \beta_1 > \frac{1}{4\left(1+\frac{u_1}{u_2}\right)}$$

wherein β_1 and β_2 denote the angles formed at the outlet from the blade ring on the working face between the peripheral direction and the base line of the blade profile (e. g. the line traced tangentially to the working face of the blade profile) of any blade ring on the casing, and of the adjacent blade ring cooperating therewith on the rotor, respectively, and u_1 and u_2 denote the corresponding peripheral velocities of these blade rings, the casing and the blades carried thereby being made rotatable in a direction opposite to the direction of rotation of the rotor.

3. In a multiple stage axial flow rotary compressor or pump comprising a rotor and a stationary casing, a plurality of blade rings on the rotor alternating with a plurality of blade rings on the casing, the mean diameter of any blade ring being at least approximately equal to the average of the mean diameters of the adjacent blade rings, the blades of each ring having airfoil profiles and being positioned so that the value of the quotient composed of the tangents of the angles β_1 and β_2 should at least on one blade diameter along the blade length fall within the limits 2 and 0, wherein β_1 and β_2 denote the angles formed at the outlet from the blade ring on the working face between the peripheral direction and the base line of the blade profile (e. g. the line traced tangentially to the working face of the blade profile) of any stationary blade ring carried by the stator, and of the adjacent rotating blade ring carried by the rotor and cooperating with the said stationary blade ring, respectively.

4. In a multiple stage axial flow rotary compressor or pump comprising a rotor and a stationary casing, a plurality of blade rings on the rotor alternating with a plurality of blade rings on the casing, the mean diameter of any blade ring being at least approximately equal to the average of the mean diameters of the adjacent blade rings, the blades of each ring having airfoil profiles and being, at least on one blade diameter along the blade length, positioned so that the value of the quotient composed of the tangents of the angles β_1 and β_2 should fall within the limits 2 and 0, besides, $\tan \beta_1$ should fall within the limits 3 and ¼, wherein β_1 and β_1 denote the angles formed at the outlet from the blade ring on the working face between the peripheral direction and the base line of the blade profile (e. g. the line traced tangentially to the working face of the blade profile) of any stationary blade ring carried by the stator, and of the adjacent rotating blade ring carried by the rotor and cooperating with the said stationary blade ring, respectively.

5. In a multiple stage axial flow rotary compressor or pump comprising a rotor and a stationary casing, a plurality of blade rings on the rotor alternating with a plurality of blade rings on the casing, the mean diameter of any blade ring being at least approximately equal to the average of the mean diameters of the adjacent blade rings, the blades of each ring having airfoil profiles and being, at least on one blade diameter along the blade length, positioned so that the angles β_1 and β_2 should be mutually equal, and $\tan \beta_1$ should fall within the limits 3 and ¼, wherein β_1 and β_2 denote the angles formed at the outlet from the blade ring on the working face between the peripheral direction and the base line of the blade profile (e. g. the line traced tangentially to the working face of the

blade profile) of any stationary blade ring carried by the stator, and of the adjacent rotating blade ring carried by the rotor and cooperating with the said stationary blade ring, respectively.

6. In a multiple stage axial flow rotary compressor or pump comprising a rotor and a casing a plurality of blade rings on the rotor alternating with a plurality of blade rings on the casing, the mean diameter of any blade ring being at least approximately equal to the average of the mean diameters of the adjacent blade rings, the blades of each ring having airfoil profiles and being positioned so that the value of the quotient composed of the tangents of the angles β_1 and β_2 should at least on one blade diameter along the blade length fall within the limits:

$$\frac{2}{1+3\frac{u_1}{u_2}} > \frac{\tan \beta_1}{\tan \beta_2} > 0$$

further, the same quotient of tangents should on the various blade diameters along the blade length at least approximately follow the law

$$\frac{\tan \beta_1}{\tan \beta_2} = qr^2 - 1$$

wherein q is a suitably selected constant and r is the distance of the various points of the blade length measured from the axis of rotation, whilst β_1 and β_2 denote the angles formed at the outlet from the blade ring on the working face between the peripheral direction and the base line of the blade profile (e. g. the line traced tangentially to the working face of the blade profile) of any blade ring on the casing, and of the adjacent blade ring cooperating therewith on the rotor, respectively, finally u_1 and u_2 denote the corresponding peripheral velocities of these blade rings, the casing and the blades carried thereby being made rotatable in a direction opposite to the direction of rotation of the rotor.

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