United States Patent [19]

Dzung

[11] **3,764,225**

[45] **Oct. 9, 1973**

[54] **TECHNIQUE AND BLADE ARRANGEMENT TO REDUCE THE SERPENTINE MOTION OF A MASS PARTICLE FLOWING THROUGH A TURBOMACHINE**

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[22] Filed: **Apr. 30, 1971**

[21] Appl. No.: **139,037**

[30] **Foreign Application Priority Data**

May 27, 1970 Switzerland 7241/70

[52] **U.S. Cl.** .... **415/193,** 415/199 R

[51] **Int. Cl.** **F01d 1/04**

[58] **Field of Search** .... 415/191, 192, 193, 415/194, 195, 199, 213 C, 210, 119

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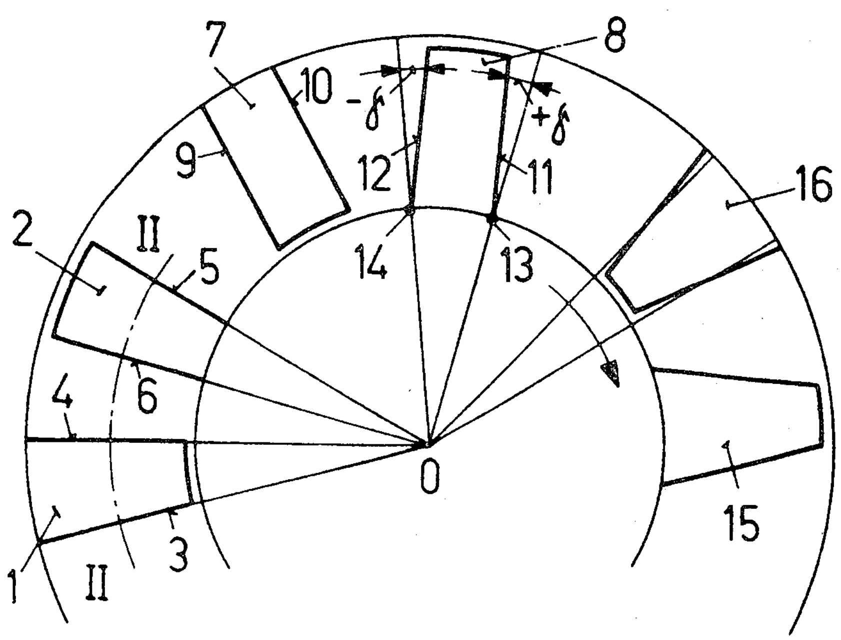
[57] **ABSTRACT**

An axial-flow turbo-machine where, in order to reduce serpentine motion of the mass particles of the working medium in flowing through the machine, the fixed and movable blades are shaped in such manner that radial forces are induced in the working medium which compensate, at least in part, the radial forces resulting from the peripheral component of the flow velocity which are responsible for the serpentine motion.

1 Claim, 5 Drawing Figures



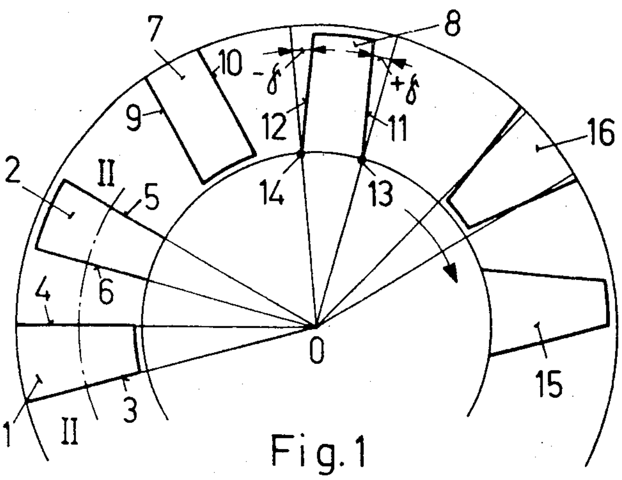


Fig. 1

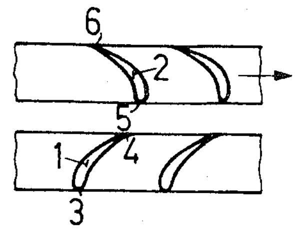


Fig. 2

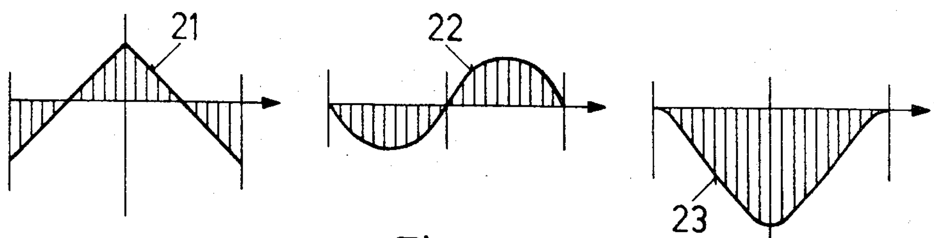


Fig. 3

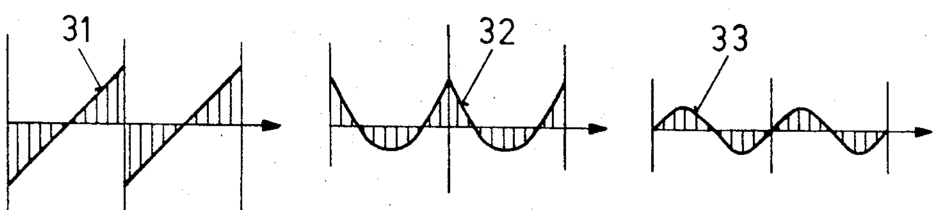


Fig. 4

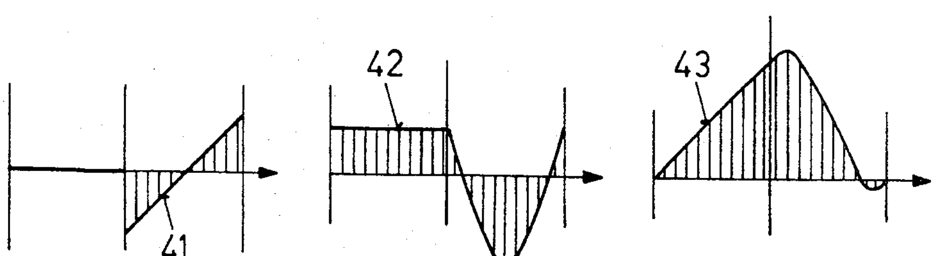


Fig. 5

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TECHNIQUE AND BLADE ARRANGEMENT TO REDUCE THE SERPENTINE MOTION OF A MASS PARTICLE FLOWING THROUGH A TURBOMACHINE

The present invention relates to an improved technique for reducing the serpentine motion, caused by the alternating peripheral component of the flow velocity, of the path, projected cylindrically on a meridional plane, of a mass particle in an axial-flow turbomachine, by means of the blading of the machine, and a blade arrangement to effect this technique.

In the blading of an axial-flow turbomachine, the flow medium is subjected to a serpentine motion in the meridional plane owing to periodic variation of the peripheral component of the flow velocity and to the associated variation of the centrifugal forces. This serpentine motion gives rise to additional energy losses and, by affecting the flow approach angles, makes correct determination of the blade profiles difficult.

A well-known countermeasure consists in twisting the blades so that the flow is free from eddies, at least in the axial direction. An eddy-free flow can satisfy the condition of radial equilibrium without serpentine motion. The disadvantage of this feature is that with machines of high volume through-put, and hence large blade-diameter ratios, the blades have to be very sharply twisted. This increases manufacturing costs and for reasons of strength is not always practicable.

Attempts have also been made to set the blades obliquely, relative to the radial direction. The radial forces were meant to compensate the centrifugal forces; this feature proved ineffective.

The purpose of the present invention is to reduce the serpentine motion of the working medium on its way through the turbo-machine, and also the energy losses thus caused. In accordance with the invention this is achieved by making the blades of such a shape that radial forces are induced in the working medium which at least partially compensate the radial forces resulting from the peripheral component.

A blade arrangement achieving this purpose is characterized by the fact that, when viewed in the axial direction and considered from base to tip of the blade, the leading and trailing edges of each fixed blade converge to at least such an extent that they coincide with a radius, and the leading and trailing edges of each moving blade, also viewed in the axial direction and considered from base to tip of the blade, deviate from the radii passing through the base points of the leading and trailing edges to at least such an extent that they are approximately parallel or even somewhat convergent.

The invention is further explained below with the aid of the appended drawings. These show:

FIG. 1 different forms of fixed and moving blades, viewed in the direction of the axis of the turbomachine,

FIG. 2 plane representation of part of cylindrical section II—II of FIG. 1, through one row each of fixed and moving blades,

FIG. 3–5 associated flow diagrams.

To aid understanding of the invention, the cause of the serpentine motion of a mass particle must be examined more closely. Consider a typical stage of a multiple-stage, axial-flow turbo-machine with 50 percent reaction along the radius (i.e., the fixed and moving blades each convert one half of the stage drop). In front of the fixed row of the stage, the peripheral component of the flow velocity has a certain value which is first considered arbitrarily to be in a negative direction. Within the blades of the fixed row the peripheral component of this negative value increases to a certain new, positive value. On flowing through the blades of the moving row the absolute peripheral component of velocity is converted into work in accordance with Euler's turbine law. On leaving the moving row, this component again has the same negative value as in front of the fixed row, whereupon this process is repeated in the next stage. This variation of the peripheral component causes corresponding variation of the radial acceleration which, with constant reaction, is proportional to this velocity component. The radial acceleration in turn gives rise to a radial component of velocity, and hence to radial compression of the streamlines in the meridional plane. This is the phenomenon of serpentine motion.

FIG. 1 and 2 show a fixed blade **1** of a turbomachine with leading edge **3** and trailing edge **4**, and also a moving blade **2** with leading edge **5** and trailing edge **6**. The leading and trailing edges of both blades coincide with radii, i.e., lines passing, through centre **0**.

Curve **21** in FIG. **3** represents schematically the radial acceleration of a mass particle in a stage as caused by the radial forces resulting from the peripheral component of the flow velocity. Curve **22** illustrates the velocity, and curve **23** the radial displacement, where in each case, as also in FIG. **4** and **5**, the left-hand half of a diagram represents conditions in the fixed row, and the right-hand half conditions in the moving row of a stage. The abscissae are in each case the time taken by a mass particle of the working medium to flow through the stage. Except for the influence of the constriction due to the blade thickness, this time is proportional to the axial distance, so that curve **23** also represents the meridional streamline projected cylindrically in a meridional plane. Curves **22** and **23** are obtained by single and double integration, respectively, of curve **21**, with the aid of suitable integration constants. The above statements require modification if account is taken of the finite thickness of the blade.

If all the blades could be made "radial," as blades **1** and **2** in FIG. **1**, then, apart from the radial acceleration caused by the flow process described above, no radial blade force could be applied to the working medium. The traditional form of fixed and moving blades, however, is "cylindrical," i.e., with practically parallel leading edges, as fixed blade **7** and moving blade **8** in FIG. **1**. The leading and trailing edges then make the angle $\pm\gamma$ with the radii passing through their base points, such that the leading edge **9** of fixed blade **7** and the trailing edge **12** of moving blade **8** form negative angles, while the trailing edge **10** of fixed blade **7** and the leading edge **11** of moving blade **8** form positive angles, as is shown for moving blade **8**, with base points **13** and **14** for leading edge **11** and trailing edge **12**, respectively. These angles γ are at the same time the angles between the radial and peripheral components of the forces exerted by the blades on the working medium. Since the peripheral components of the flow velocity in the fixed and moving blades are always in mutually opposed directions, the additional radial blade-force components in the fixed and moving blades, and hence also the additional radial accelerations, always follow a similar course, e.g. as curve **31** in FIG. **4**. Integration of curve **31** again yields velocity curve **32** and displacement curve **33.**

Comparison of FIG. **3** and **4** shows that the radial forces in the fixed blades are added to each other, although they compensate each other in the moving blades. This applies to both a turbine and a compressor. As in the case of an oscillation of different frequencies the two causes cannot cancel out each other. Indeed, the dissipation loss will always be cumulative.

In accordance with the invention, the fixed blades are so formed that their leading edges and trailing edges are essentially radial as indicated by blade **1** in FIG. **1** so that there are no radial forces acting on the fluid. The moving blades, e.g., blade **8** in FIG. **1,** are so formed that the leading edges, e.g., edge **11,** are inclined at a positive angle of $+\gamma$ with respect to the radial line so that the concave side of each blade, i.e., the pressure side, as represented by the left-hand side of the moving blade in FIG. **2** is slightly facing toward the axis, while the trailing edge, e.g., edge **12** of blade **8** is inclined at an opposite, i.e., a negative angle of $-\gamma$ against the radial line. In this way, there exists a component of force acting by the moving blades on the flowing fluid in the direction towards the center at the entrance and away from the center at the outlet. Curve **41** in FIG. **5** shows these forces acting on the fluid or the acceleration within the stage. By the acceleration within the stage. By comparing with FIG. **3** it is evident that the radial forces due to the peripheral component of the flow velocity and to the shape of the blading compensate each other in the moving blades, but that no additional disturbing force occurs in the fixed blades. Integration with the appropriate integration constants yields curve **42** for the velocity and curve **43** for the displacement. Comparison of curve **43** with curve **23** of FIG. **3** shows that suitable choice of the relationship between the two causes of radial displacement results in very effective compensation.

The effect of compensation can be increased if the leading and trailing edges of the moving blades viewed in the axial direction and considered from base to tip of the blade converge as shown by **15** in FIG. **1.** In this way the relationship between the two causes of radial displacement of a mass particle can be altered. It must be expressly emphasized here that this blade form is not to be confused with the well-known technique of "tapering" the blades. This is, as a rule, such that in axial section the blades are indeed broader at the base than at the tip, but that the relationships in the peripheral direction are reversed.

Also in the case of the fixed blades a compensating force can be applied if the leading and trailing edges, considered from base to tip of the blade, are inclined towards each other even more than the corresponding radii, e.g., as **16** in FIG. **1.** If the radial force in the fixed blades acts in the appropriate direction, almost complete compensation can be achieved.

A similar increase in effectiveness can be achieved with a blade arrangement in accordance with the invention such that the moving blades are broader than the fixed blades in either the peripheral direction or the axial direction. As comparison between FIGS. **3** and **5** has shown, a compensating effect occurs only in the case of the moving blades. When the moving row is wider in the axial direction, the time taken in flowing through the moving blades is greater; the same radial acceleration then gives rise to a higher value of the radial velocity and of the radial displacement. If the moving blades are broader in the peripheral direction, the value of angle $\pm\gamma$ in FIG. **1** is higher, thus reinforcing the compensating effect.

I claim:

1. A multi-stage axial-flow turbo-machine, each stage thereof comprising a row of fixed blades having a concave curved configuration and an adjacent row of movable blades having an oppositely concave curved configuration, the leading and trailing edges of said fixed blades of each stage extending along radial lines as viewed in the direction of the axis, the leading edges of said movable blades of each stage being inclined at a positive angle with respect to a radial line whereby the concave side of each movable blade faces slightly towards the axis, and the trailing edges of said movable blades of each stage being inclined at the same but negative angle with respect to a radial line, thereby to effect a reduction in the serpentine motion of a mass particle as it passes through successive stages of said multistage turbo-machine and which is caused by the alternating peripheral component of the flow velocity of the particle path projected cylindrically on a meridonal plane.

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