

SPECIFICATION

Improvements in or relating to regenerative rotodynamic machines

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This invention relates to regenerative rotodynamic machines, and more especially to regenerative pumps and compressors.

- A regenerative or peripheral pump is a
 10 rotodynamic machine which permits a head equivalent to that of several centrifugal stages to be obtained from a single rotor with comparable tip speeds. The impeller can take the form of a disc with a set of vanes projecting axially at each side near the disc periphery.
 15 Around the greater portion of the periphery the vanes project into an annular channel of which the cross sectional area is greater than that of the impeller vanes. At one sector
 20 between the inlet and discharge the annular channel is reduced to a close running clearance around the impeller. This sector is called the stripper seal and its function is to separate the inlet and discharge ports, thereby forcing
 25 the fluid out through the discharge port. The stripper allows only the fluid between the impeller vanes to pass through to the inlet.

- The advantage of pumps of this type lies in the generation of a high head at low flow rates. They have a very low specific speed. Although their efficiency is not very high, being usually less than 50%, pumps of this type have found many applications in industry where it is preferred to use rotodynamic
 35 pumps in place of positive displacement pumps for duties requiring a high head at low flow rates. Their simplicity, and the absence of problems due to lubrication and wear, give advantages over positive displacement pump,
 40 despite the lower efficiency.

- The regenerative pump has been adapted for the compression of gas. The advantage lies in the low specific speed giving a high pressure ratio together with a low flow rate for
 45 a given size of machine. Further advantages are oil free operation and freedom from stall or surge instability.

- In such a compressor, the gas follows a helical path through the annular channel and
 50 passes through the vanes a number of times in its peripheral path from the inlet port to the discharge port. Each passage through the vanes may be regarded as a stage of compression and thus the equivalent of several stages
 55 of compression can be obtained from a single impeller. This pumping process, however, cannot be considered as efficient. The fluid between the vanes is thrown out and across the annular channel and violent mixing occurs, the angular momentum acquired by the
 60 fluid in its passage between the vanes being transferred to the fluid in the annular channel. The mixing process is accompanied by the production of a great deal of turbulence and
 65 this implies an undesirable waste of power.

Several theories of the fluid-dynamic mechanism of a regenerative pump have been published. These theories have been reviewed and compared by Senoo (A.S.M.E. Trans. Vol. 70 78, 1956, pp. 1091-1102). Differences occur in the assumptions made, but in principle the various theories appear to be compatible. Senoo and Iversen (A.S.M.E. Trans. Vol. 77, 1955, pp 19-28) consider turbulent friction
 75 between the moving impeller and the fluid as the primary force causing the pumping action. Wilson, Santalo and Oelrich (A.S.M.E. Trans. Vol. 77, 1955, pp 1303-1316) regard the mechanism as based on a circulatory flow
 80 between the impeller and the fluid in the casing with an exchange of momentum between the fluid passing through the impeller and the fluid in the casing.

More recently, compressors with considerably better efficiency have been proposed in which the conventional radial vanes are replaced by aerodynamic blading. The annular channel is provided with a core to assist in guiding the fluid so that it circulates through
 90 the blading with a minimum of loss. The core also acts as a shroud closely surrounding the blades at their tips to reduce losses due to the formation of vortices at the tips of the blades. Such an arrangement is described, for instance, in British Patent Specification No. 1237363.

It is an object of this invention to achieve further important improvements in regenerative rotodynamic machines having aerodynamic
 100 blading.

According to the present invention, there is provided a fluid dynamic machine including a rotor, a peripheral part of which is provided with a set of blades; and a stator including
 105 structure defining an annular channel accommodating said peripheral part, the walls of said channel being spaced from the set of blades to form an open toroidal passage extending angularly around a major portion of
 110 said periphery between inlet and outlet ports of the stator and the remaining angular extent of said channel having minimum clearance about said blades, said blades being curved to give a turning angle in the plane containing
 115 the curvature of the blades of substantially more than 90° to fluid flow through the set of blades in use, fluid transmitted through said passage coacting with said blades to follow a spiral path entering and leaving said set of
 120 blades more than once as the rotor rotates in use.

In Oxford University Engineering Laboratory Report No. 1.139/75, Dr. H. Sixsmith (one of the inventors of the present application) and
 125 his co-worker Dr. H. Altmann proposed aerodynamic blades with a turning angle of 90°; at the time this was thought to be the maximum usable angle. However, we have now discovered that there is considerable advantage
 130 in increasing the angle to well above

90°.

An arrangement of compressor in accordance with the invention will now be described, by way of example, with reference to the accompanying drawings, in which:—

Figure 1 is a diagrammatic cross-section of a regenerative compressor according to the invention,

Figure 2 is a diagram of the aerodynamic blade profile, and

Figure 3 is a diagrammatic representation of the blade velocities and flow angles.

Referring firstly to Fig. 1, this shows diagrammatically a simple single impeller regenerative compressor according to the invention. The impeller 11 housed in a split casing 25 is driven by a shaft 10 and consists of a disc with aerodynamic blades 18A, 18B provided within scooped out regions 12A, 12B at each side of the disc just radially inward of the disc periphery. The bladed margin of the impeller projects into an annular chamber 13 in the compressor casing 25 which is wider than the impeller and has at its outer periphery an inward-facing cylindrical surface 14 which is closely approached by the cylindrical peripheral surface 15 of the impeller 11, thereby dividing the chamber 13 into two separated side channels 13A, 13B, each of roughly oval cross-section, that are located on opposite sides of the impeller disc 11 and are each defined partly by the wall of the chamber 13 and partly by the contour of the respective scooped out side portion 12A or 12B of the impeller 11 that contains the blades 18A or 18B. The blades extend approximately half-way across the respective side channel 13A, 13B and are designed to turn the fluid through an angle of well in excess of 90° as it flows radially outward through an angle of well in excess of 90° as it flows radially outward through the blading, setting up a circulation in each side channel 13A, 13B as indicated by the arrows F. Each annular side channel has a central core 16A, 16B to assist in guiding the fluid so that it circulates through the blading with a minimum of loss. Each core 16A, 16B is in the form of a shroud ring placed against the blade tips to eliminate loss due to formation of vortices at the tips of the blades.

The fluid enters the annular chamber 13 through a port 19 in the wall of the casing 25 which leads to an inlet chamber 20 communicating with both of the channels 13A, 13B at their outer peripheries. The fluid leaves the annular channels 13A, 13B through an outlet (not shown) which is followed by a conical diffuser to obtain pressure recovery. Between the inlet and outlet, the stripper seal (not shown) is formed by shaping the interior of the casing walls so that they approach closely to the sides of the impeller all the way out to its periphery 15. Alternatively, the stripper seal can be formed by the addition of a

completely separate stripper element.

Radially inward of the scooped cavities 12A, 12B and blading 18A, 18B, the impeller 11 is formed as an annular dish, with a hollow interior 23 closed by an annular plate 27.

Between the inlet and outlet ports, the fluid being compressed passes a number of times through the blading 18A, 18B. During each passage a quantity of energy is transferred from the impeller to the fluid. The rate of flow through the blading is self-adjusting in the sense that the velocity through the blade channels tends to increase until the rate of energy transfer reaches the value needed to generate the pressure difference between the inlet and outlet ports. An increase in the pressure difference causes corresponding increases in both the number of passages through the blading and the energy transferred at each passage. The rate of energy transfer tends to vary as the square of the velocity relative to the blades. By equating the power transferred from the blading to the fluid with the power needed to generate the pressure difference across the inlet and outlet ports, the flow velocities in the annular channels 13A, 13B can be estimated. This information serves as a useful guide towards the optimum design of the blading.

Referring to Fig. 3, it is seen that the fluid enters and leaves the blading with relative velocities W_1 and W_2 and with inlet and outlet fluid angles of β_1 and β_2 . If V_{U1} and V_{U2} are, respectively, the peripheral components of the absolute velocities of the fluid at the leading and trailing edges of the blading, and U_1 and U_2 are the peripheral velocities of the leading and trailing edges, then:—

$$V_{U1} = U_1 - W_1 \sin \beta_1$$

$$V_{U2} = U_2 + W_2 \sin \beta_2$$

The peripheral or forward component of velocity of the gas on leaving the blades is greater than the blade velocity. As soon as the gas emerges from the blades, it comes under the influence of the peripheral pressure gradient and during its transverse passage around the annular channel its peripheral velocity is progressively reduced until it re-enters the blading to receive another impulse. As seen in Fig. 2, for ease of manufacture the surfaces of the aerodynamic blades 12A, 12B are formed of successions of circular arcs. In the example illustrated, the inner surface 40 of the blade is formed as a single arc while the outer surface 31 is formed as a central 80° arc flanked by two 15° arcs and then two 18° arcs.

Machines according to the invention are balanced and vibration free and, being comparatively inexpensive to build, provide a quieter alternative to the Roots blower. Existing regenerative compressors are equally smooth running but not so efficient. Thus, such prior

machines give a maximum of 8 p.s.i. in one stage whereas machines according to the invention will give 10 p.s.i. and upwards, and also can be employed to pull a vacuum.

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CLAIMS

1. A fluid dynamic machine including a rotor, a peripheral part of which is provided with a set of blades; and a stator including structure defining an annular channel accommodating said peripheral part, the walls of said channel being spaced from the set of blades to form an open toroidal passage extending angularly around a major portion of said periphery between inlet and outlet ports of the stator and the remaining angular extent of said channel having minimum clearance about said blades, said blades being curved to give a turning angle in the plane containing the curvature of the blades of substantially more than 90° to fluid flow through the set of blades in use, fluid transmitted through said passage coacting with said blades to follow a spiral path entering and leaving said set of blades more than once as the rotor rotates in use.

2. A machine as claimed in claim 1, wherein said blades have an aerofoil cross-section.
3. A machine as claimed in claim 2, wherein said blades have a concave leading face and a convex trailing face of substantially different radii of curvature.

4. A machine as claimed in claim 3, wherein the leading face is of uniform curvature and the trailing face is of compound curvature increasing in radius laterally towards entry and exit lips of the blades.

5. A machine as claimed in any of the preceding claims, wherein each blade is symmetrically curved to each side of a centre line of the blade.

6. A machine as claimed in any one of the preceding claims, wherein the radial height of the blades is greater than their axial width.

7. A machine as claimed in any one of claims 1 to 5, wherein the blades extend axially of the rotor.

8. A machine as claimed in any one of the preceding claims, including an annular shroud mounted in the centre of the toroidal passage and at the outer edges of the rotor blades.

9. A machine as claimed in any one of the preceding claims, wherein the rotor includes a pair of sets of blades, each set being accommodated in a respective annular channel of the stator.

10. A machine as claimed in claim 9, wherein the sets of blades extend axially of the rotor on opposite sides of a peripheral part thereof.



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