

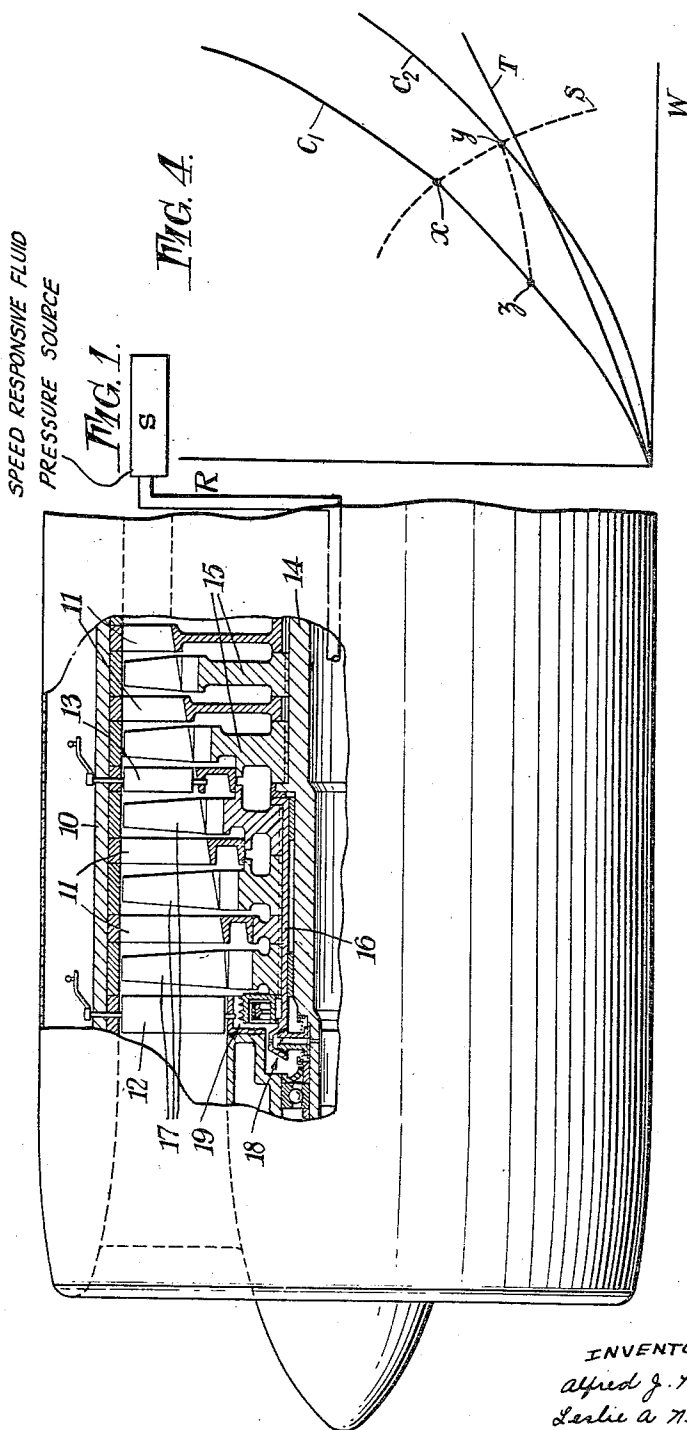
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AXIAL FLOW COMPRESSORS

2,820,588

Filed Dec. 11, 1953

2 Sheets-Sheet 1



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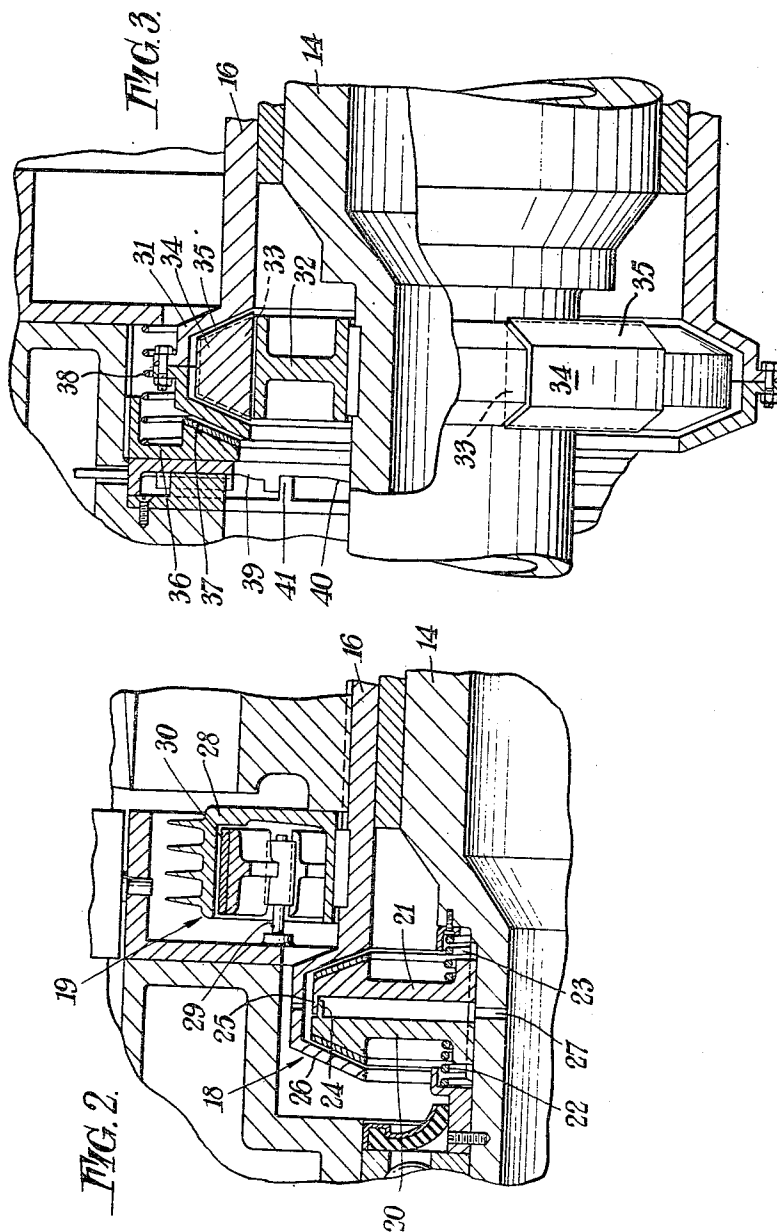
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AXIAL FLOW COMPRESSORS

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5 Claims. (Cl. 230—114)

This invention relates to multi-stage axial flow compressors such as are used, for instance, for compressing the working medium in gas turbine plants.

It is an object of the present invention to provide a multi-stage axial flow compressor which will facilitate starting and acceleration and which has an improved part-load performance.

In a multi-stage axial flow compressor according to the present invention, a part of the rotor which carries blades of the earlier stages is mounted so that it is capable of rotating independently of another part of the rotor which carries blades of the later stages and to which the drive is imparted, and a disengageable coupling is provided for connecting the rotor parts together.

The reason why this arrangement can facilitate starting and acceleration is as follows. A compressor rotor has a large inertia, and it must be run up to quite a high speed before the plant becomes self-sustaining, that is to say, when the turbine is producing sufficient power to drive the compressor without external aid. It is therefore necessary for the starting motor to impart a large amount of energy to the rotor. In the present invention, by disengaging the coupling for starting, the inertia of the rotor is reduced in that the starting motor only has to accelerate the part of the rotor which carries the blades of the later stages. When the plant has become self-sustaining, or when the turbine is producing enough power to reduce the load on the starting motor, the coupling can be engaged so that the part of the rotor which carries the blades of the earlier stages will be speeded up to the same speed as the positively driven part and the two parts will then rotate as one. Thereafter the compressor will act in the same way as an axial flow compressor of orthodox construction. When the coupling is disengaged, the blades of the earlier stages will offer only slight resistance to the flow of working medium induced by the later stages, so that the only adverse effect which the earlier stages have is to produce a slight pressure drop in the working medium which transverses them.

As indicated above, the advantages of the invention are not confined to the improvement in starting and acceleration, since an improved part-load performance is also obtainable.

In a gas turbine plant it would be desirable for the pressure-mass flow characteristic of the compressor to match that of the turbine over the whole range of mass flow. Unfortunately, the characteristics of conventional axial flow compressors and turbines are such that, if they are properly matched for normal full-load conditions, they are not properly matched at part-load, with the result that the compressor operation may become unstable. One way of dealing with the problem is to vary the compressor characteristic by providing it with adjustable guide vanes. However, such vanes may not be sufficiently effective at medium or small mass flows, with the result that medium or low load running is apt to be unstable.

By the present invention, at medium or low loads, the

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earlier stages can be rendered inoperative so that, in effect, the compressor is converted into one of fewer stages, with a better surge characteristic in the medium and lower part of the mass flow range. When the mass flow has increased to an appropriate value, the first stages may be brought into operation by engaging the coupling, thereby converting the compressor, in effect, to one of conventional type.

The coupling together of the compressor parts at the desired speed may be effected automatically, for instance by a centrifugal coupling adapted to couple the two parts together at a predetermined rotational speed.

In one form of the invention adjustable guide vanes are provided for operation in the intermediate ranges. This has the advantage that it enables the coupling to be engaged at a lower speed than would be the case if no such vanes were provided.

A second disengageable coupling may be provided between a stationary part of the compressor and the part of the rotor which carries the blades of the earlier stages, so that this part of the rotor can be held stationary when the first-mentioned coupling is disengaged. Then at an appropriate speed, the second coupling can be disengaged and the first-mentioned coupling engaged, so that the part of the rotor carrying the blades of the earlier stages is rapidly accelerated from rest until it is rotating at the same speed as the second part of the rotor. If desired there may be a delay between the disengagement of the second coupling and the engagement of the first coupling, to permit the initial acceleration of the part of the rotor carrying the blades of the earlier stages to be accomplished by the "windmill" effect of the air flow through these blades.

As an alternative to holding stationary the part of the rotor which carries the blades of the earlier stages, this part can be permitted to "windmill" at low mass flows, in which case the second coupling may be dispensed with.

Where two disengageable couplings are provided, these may be combined in a single mechanism, i. e. mechanism in which at least one part is common to both couplings.

The invention may be performed in various ways, and one particular embodiment and a modification thereof will be specifically described by way of example with reference to the accompanying drawings, in which:

Figure 1 is a side elevation of the axial flow compressor which is partially cut away and sectioned to show the internal arrangements;

Figure 2 shows a part of the mechanism of Figure 1 on a larger scale;

Figure 3 shows another form of mechanism which is alternative to that shown in Figure 2; and

Figure 4 is a family of curves illustrating the performance of the compressor.

The axial flow compressor shown in Figure 1 is intended for supplying compressed air to a turbine (not shown) via a combustion chamber (not shown). In the compressor there is a cylindrical casing 10 which carries several rings of fixed stator blades 11 and two rings 12 and 13 of angularly adjustable guide vanes. Running through the centre of the compressor is a shaft 14 on which are fixed several rotor blade rings 15 which comprise the later stages of the compressor. Mounted on the shaft 14 so as to be freely rotatable thereon is a sleeve 16 on which are mounted several rotor blade rings 17 which comprise the earlier stages of the compressor. The sleeve 16 can be clutched to the shaft 14 by a disengageable coupling 18. Alternatively, it can be locked to a stationary part of the compressor by another disengageable coupling 19. The couplings 18 and 19 are shown in more detail in Figure 2.

Dealing first with the coupling 18; this comprises two cone discs 20 and 21 which are splined to the shaft 14

and are urged together by springs 22 and 23. Sealing flanges 24 and 25, which can slide over each other, are provided near the peripheries of the cone discs 20 and 21 respectively. Mounted on the end of the sleeve 16 is an annular channel 26 of trough-shaped cross-section, the inclination of its sides being the same as the inclinations of the cone surfaces of the discs 20 and 21. To the space between the discs 20 and 21 which is bounded on its outer periphery by the sealing flanges 24 and 25, a fluid under pressure (e. g. oil or air) can be admitted from the hollow centre of the shaft 14 through a passage 27. For automatic operation, this fluid pressure may be a function of the speed of rotation, for instance air at the delivery pressure of the compressor or oil supplied by a centrifugal pump, such as speed responsive fluid pressure source S as shown in Figure 1. When fluid under pressure is so admitted, the cone discs 20 and 21 are forced apart against the action of springs 22 and 23 so that their cone surfaces engage the walls of the channel 26 and thus lock the sleeve 16 to the shaft 14. Conversely, when the pressure is released, the springs 22 and 23 force the cone discs 20 and 21 out of engagement with the trough member 26 so that the sleeve 16 is freed from the shaft 14 in the rotational sense.

Turning now to the coupling 19; there is keyed to the shaft 16 a drum 28. Mounted within the drum 28 on fixed pins 29 are arcuate friction shoes 30 which, by means of an expanding mechanism (not shown), can be forced outwardly to engage the drum 28 and thus hold this drum stationary, and with it the sleeve 16.

In the arrangement shown in Figures 1 and 2, the disengageable couplings 18 and 19 are actuated independently by suitable control mechanisms and can be brought into operation as and when required to suit the running conditions. However, it is possible to substitute for the disengageable coupling 18 shown in Figures 1 and 2 an automatic clutch mechanism which comes into operation to lock the sleeve 16 to the shaft 14 when the rotational speed of the latter has risen to a certain value. Figure 3 shows such a mechanism. In this mechanism the end of the sleeve 16 carries an annular channel 31 of trough-shaped cross-section, which is similar to the channel 26 of Figure 2. Keyed to the shaft 14 is a spider member 32 which has several wide parallel-walled radial slots, the walls 33 of two of these slots being indicated in Figure 3. In these slots are weights 34 which can slide in and out towards and away from the axis of the shaft 14. These weights have inclined flanks 35, the angle of which is the same as the angle of the trough-shaped annular channel 31. As the shaft 14 rotates, these weights fly outwards and engage the walls of the channel 31. As the speed of rotation of the shaft 14 increases, the centrifugal force exerted by these weights also increases until the frictional force which they exert on the channel 31 exceeds the resistance to movement of the sleeve 16, whereupon this sleeve begins to rotate. At normal running speeds, the centrifugal force exerted by the weights 34 is so great that there is no slip between them and the walls of the channel 31.

This mechanism also includes a somewhat different kind of disengageable coupling corresponding to the coupling 19 of Figure 2. In this case the drum 28 of Figure 2 is replaced by an annulus 36 having a friction surface 37 which can engage the outer surface of the annular channel 31. Normally this annulus is held in the "off" position by a spring 38. To the left of the annulus 36 there is an annular face-cam 39 having ramp surfaces 40. Co-operating with these ramp surfaces are fixed projections 41, so that if the face-cam is rotated, the ramp surfaces 40 will ride up on the projections 41, thereby moving the annulus 36 into frictional engagement with the outer surface of the channel member 31 and holding the sleeve 16 stationary. In this arrangement both the couplings are combined in a single mechanism in that the part 31 is common to both.

For starting the compressor the following procedure is followed. First, the sleeve 16 is held stationary by engaging the disengageable coupling 19 (expanding the shoes 30 against the drum 28 in Figure 2, or advancing the annulus 36 to the right to engage the channel 31 in Figure 3). Next, the shaft 14 is run up to speed by a starter motor (not shown). Since the sleeve 16 and the blades 17 which it carries are not rotated by the starter motor the load on this motor is less than it would be if it had to rotate the whole rotor. Rotation of the later rings or blades 15 produces some compressed air for the turbine, and at a certain speed the power generated by the turbine and available for driving the compressor will exceed the power required to rotate the shaft 14. Accordingly the compressor will continue to accelerate. The disengageable coupling may now be disengaged so that the sleeve 16 is free to rotate. The rush of air through the blades 17 starts to rotate the sleeve 16; in other words it begins to "windmill." When the sleeve is rotating fast enough it can be clutched to the shaft 14 so that the whole rotor rotates as one. In Figure 2 this clutching-in is done by applying fluid pressure between the cone discs 20 and 21 such as from speed responsive fluid pressure source S so that they grip the channel 26, while in Figure 3 the design is such that the frictional force between the weights 34 and the channel 31 at the appropriate rotational speed of the shaft 14 overcomes the drag of the sleeve 16. In both cases, the frictional drag of this disengageable coupling may be used to augment or to supplant the "windmill" effect in running the sleeve 16 up to speed.

The advantages of the invention for part-load operation are illustrated in Figure 4, which is a curve of pressure ratio (R) plotted against mass flow through the compressor (W). The curve T is the turbine load line, that is to say, it represents the mass flow of air demanded by the turbine at different pressures for a constant load. The curve C₁ represents the maximum compressor performance with only the later stages running, while the curve C₂ represents the maximum compressor performance with all stages in operation.

Assuming now that the compressor is running with only the later stages working, i. e. on the line C₁. It is clear from Figure 4 that the compressor can supply the turbine adequately on this line, but that at high speeds the pressure ratio corresponding to the desired mass flow is high, and consequently the compressor will absorb a lot of power. The power consumption can be considerably reduced at high speeds by bringing in the earlier compressor stages so that the machine works on the line C₂. At low speeds, however, the line C₂ is below the turbine load line T, so that at these speeds the output of the compressor is inadequate. Consequently, it is desirable to work on the line C₁ up to the point x, say, and then change over to the line C₂ at the point y. It will be observed that, since the change-over occurs at a definite speed, both the points, x and y lie on the same speed line S. In order to reduce the compression work still further, the compressor could be run up to a lower speed on the line C₁, say to the point z, and then from z to y the adjustment of the delivery could be effected by suitably adjusting the guide vanes 12 and 13.

What we claim as our invention and desire to secure by Letters Patent is:

1. A multi-stage axial flow compressor for compressing a gaseous medium comprising a first rotor part, blades for the first compression part mounted on said first rotor part, a second rotor part, driving means for imparting rotation to said second rotor part, means supporting said first rotor part for rotation independently of said second rotor part and said driving means, blades for the later compression stages mounted on said second rotor part, passage means which convey all the gaseous medium passed through said first compression part to said later compression stages, disengageable coupling means be-

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tween said second rotor part and said first rotor part which, when engaged, couple the said rotor parts for rotation together at the same speed, and speed responsive means for engaging said coupling means when said later stage attains a predetermined speed whereby windmilling of said first compression part is stopped and said first compression part is driven with the later stages.

2. A multi-stage axial flow compressor according to claim 1 in which said disengageable coupling means comprises a frictional coupling means having friction members, said speed responsive means including means for moving said friction members into engagement on the application of fluid pressure to one of said members, means admitting fluid pressure to said last-mentioned member, and means for regulating said fluid pressure as a function of the speed of rotation of said driving means.

3. A multi-stage axial flow compressor according to claim 1 in which said disengageable coupling means is a centrifugal coupling adapted to couple said first rotor part to said second rotor part automatically when said second rotor part is rotated above a predetermined speed.

4. A multi-stage axial flow compressor comprising a first rotor part, blades for the earliest compression stages

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mounted on said first rotor part, a second rotor part, driving means for imparting rotation to said second rotor part, means supporting said first rotor part for rotation independently of said second rotor part and said driving means, blades for the later compression stages mounted on said second rotor part, and disengageable coupling means between said second rotor part and said first rotor part which, when engaged, couple the said rotor parts for rotation together and a stationary part and disengageable coupling means between said first rotor part and said stationary part.

5. A multi-stage axial flow compressor according to claim 4 in which there is at least one member which is common to both said disengageable coupling means.

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