

April 8, 1947.

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2,418,801

INTERNAL COMBUSTION TURBINE PLANT

Filed Jan. 13, 1944

4 Sheets-Sheet 1

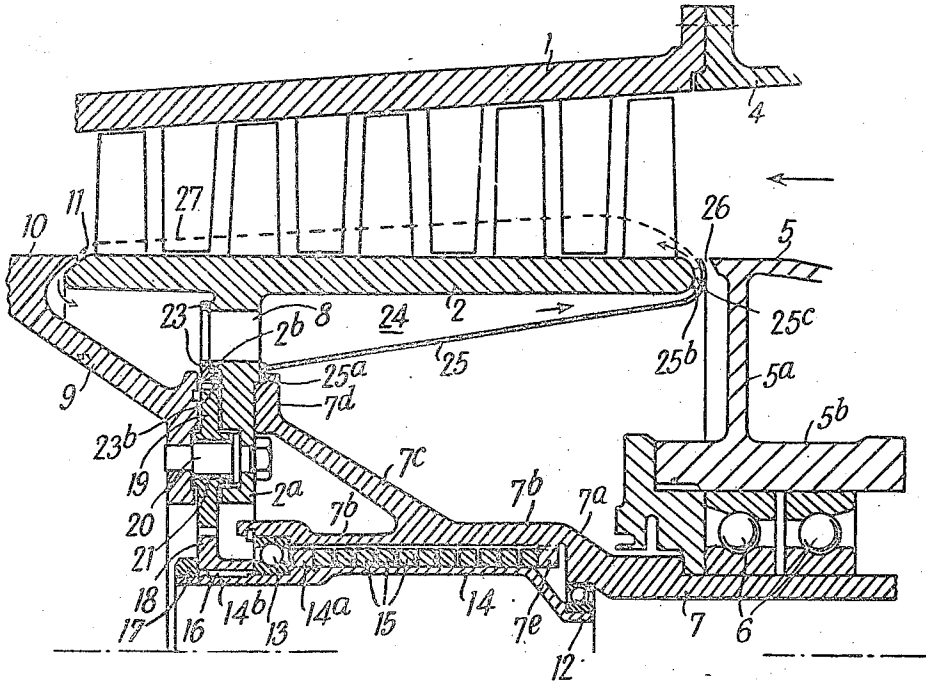


FIG. 1.

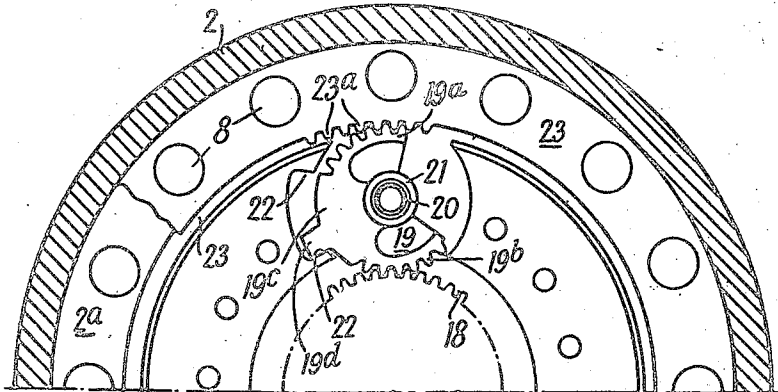


FIG. 2

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4 Sheets-Sheet 2

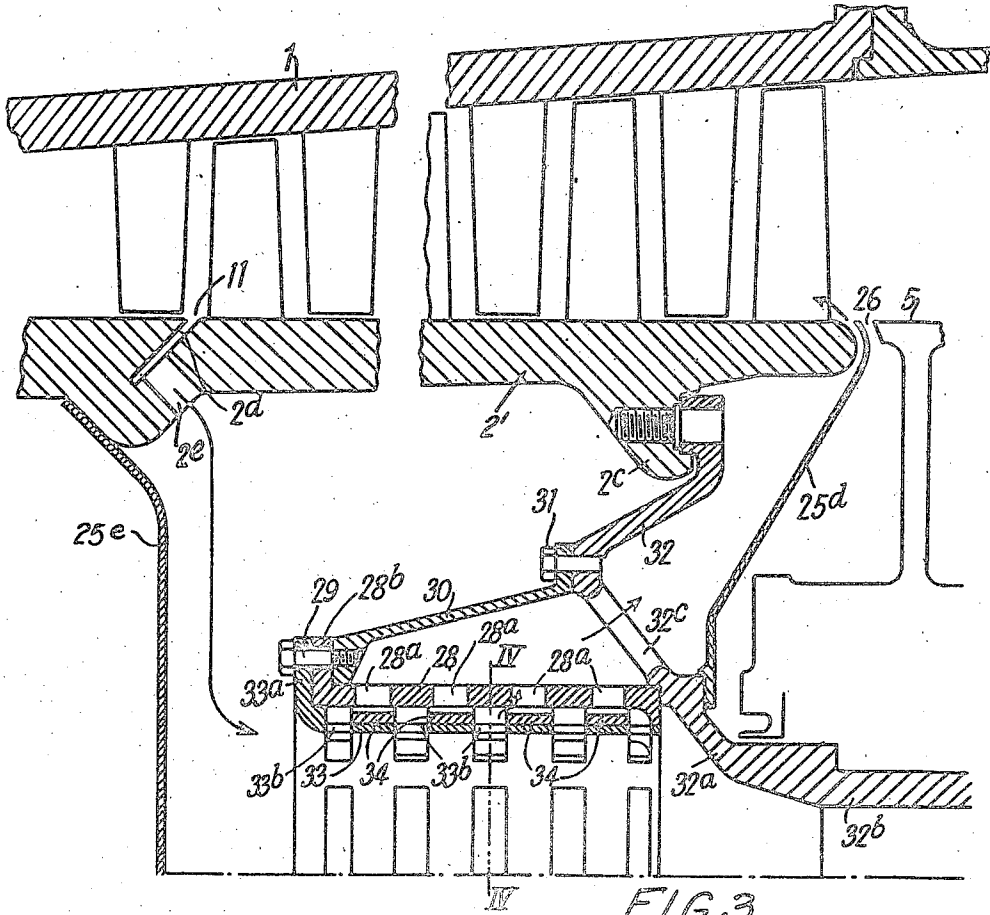


FIG. 3.

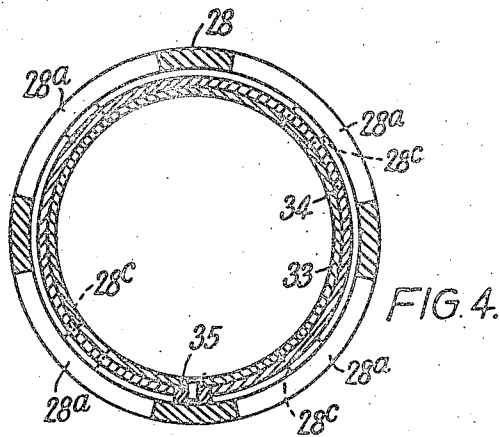


FIG. 4.

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4 Sheets-Sheet 3

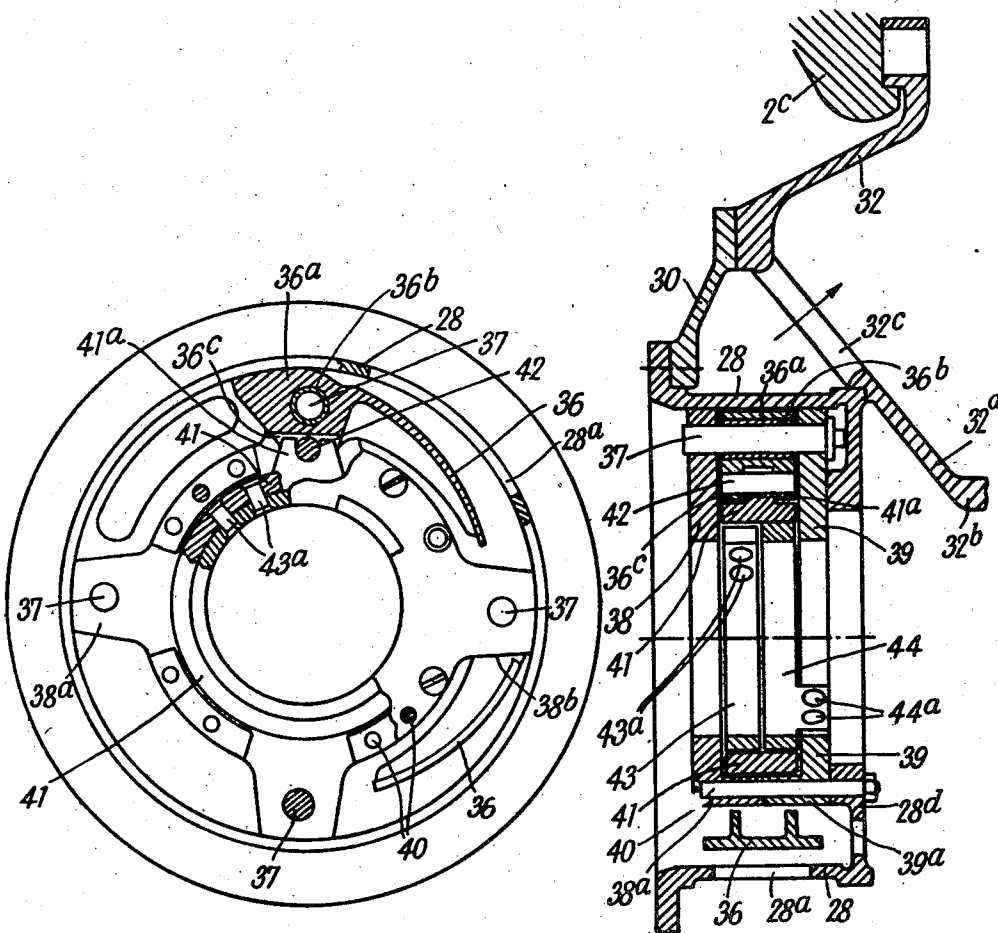


FIG. 6.

FIG. 5.

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4 Sheets-Sheet 4

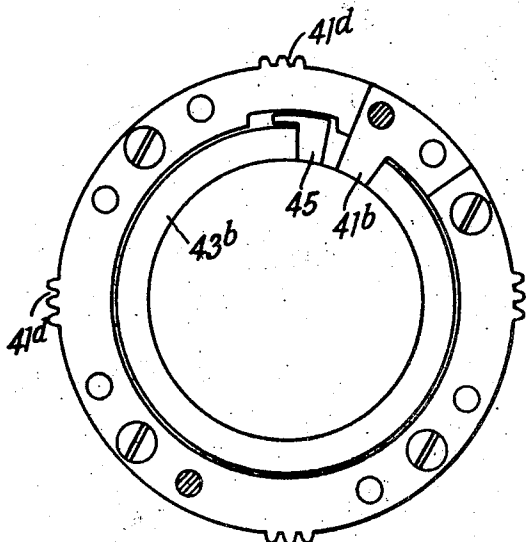


FIG. 8.

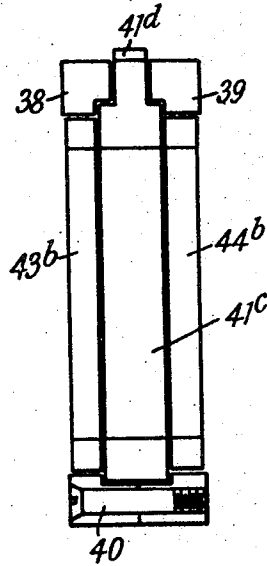


FIG. 7.

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2,418,801

INTERNAL-COMBUSTION TURBINE PLANT

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Application January 13, 1944, Serial No. 518,175
In Great Britain March 25, 1942

14 Claims. (Cl. 230-114)

1 This invention relates to an internal combustion turbine plant, hereinafter called gas turbine plant, wherein the turbine drives a multi-stage axial flow air compressor, either directly or through gearing, so that the speed of the compressor is always the same as or is in direct relation to that of the turbine; said compressor, which comprises low and high pressure parts each having a plurality of blading stages, delivers to a combustion chamber, from which the products of combustion are fed to the turbine. To obtain the requisite efficiency of such plant both the turbine and the compressor, with respect more particularly to the blading thereof, are designed to run at a predetermined speed so that a difficulty arises when the turbine and compressor run at speeds lower than the optimum speed for which they are designed. An axial flow air compressor operates satisfactorily at a predetermined speed over a very limited range of air quantity. At quantities higher than the normal the delivery pressure drops rapidly, whilst at air quantities somewhat lower than the normal the compressor becomes unstable, due to "stalling," resulting in a considerable drop in the delivery pressure and in efficiency. Furthermore, the turbine is designed so that the quantity of air, together with products of combustion which it can deal with as received from the compressor via the combustion chamber, is within the range of efficient operation of the compressor at substantially the normal speed. When, however, the speed is reduced more and more below normal the "swallowing" capacity of the turbine is reduced at a greater rate than is the natural output of the compressor, with the result that below a given speed the plant ceases to function satisfactorily, due to the stalling of the compressor. This stalling of the compressor at the lower speeds is initiated in the blading at the inlet end of the compressor, since at such lower speeds the compression ratio, and consequently also the ratio of air volumes at the inlet and outlet ends is reduced. The area of the low pressure blading is too great for the amount of air passing through the blading at the high pressure end of the compressor and through the turbine. These difficulties per se could be alleviated by making provision for the opening of additional gas nozzles in the turbine, but in practice this provision is difficult to carry out; or provision may be made for blowing off some air at the delivery end of the compressor, but this provision is conducive to inefficiency of operation. A more satisfactory method of dealing with the aforesaid difficulty

2 is to blow off some of the air from an intermediate point of the compressor, conveniently at a point between the high pressure end of the low pressure section of the compressor and the low pressure end of the high pressure section of the compressor, whereby the lowest pressure compressor blades may deal with a larger quantity of air than the highest pressure blades of the compressor and the turbine. This method, moreover, can be readily carried out as previously proposed in relation to the stationary casing of the compressor by the provision therein of valved ports which may be opened to various extents, either by hand or automatically, when the speed drops below a predetermined value. However, when the air compressor is of the contrarotational type, some examples of which per se form the subject matter inter alia of U. S. applications Serial Nos. 518,168 and 518,170, filed of even date herewith, this method cannot well be applied to the outer casing, since it is rotating. The present invention is more particularly, but not exclusively, applicable to contrarotational air compressors.

25 According to the invention, with the above considerations in view, automatically operating means are provided on the inner rotating casing of the compressor whereby air may be blown or taken off from an intermediate stage thereof, either simply to atmosphere or preferably through a duct or ducts leading to the air inlet end of the compressor. These means may be provided either alone, or in addition to means provided on the outer casing for blowing off air from an intermediate stage of the compressor when the outer casing and associated rows of blades constitute a fixed structure, and it will be appreciated that the air drawn off, according to the present invention will be in greater quantity than that which may be withdrawn in per se otherwise known manner solely for the removal of "boundary layers."

30 In carrying out the invention according to the preferred arrangement there is in the rotating inner blade-carrying drum or cylinder of the compressor at the intermediate stage thereof a port or plurality of ports through which the excess air obtaining at the intermediate stage at the lower speeds can pass to atmosphere, or through the aforesaid duct, and in either case through one or more valved ports the opening of which can be controlled by one or more valve members in accordance with the speed of rotation of the compressor. The one or more valve

35 40 45 50 55 the port or ports is or are normally fully open at

the low speeds and adapted to be automatically gradually closed as the speed increases, and to be fully closed at a speed at which satisfactory operation of the plant can take place. To this end one or more valve members may be spring-biased to the fully open position and closed by means of devices of the centrifugal governing type which may assume the forms hereinafter described.

There may be a single valve member co-operating with a plurality of ports or there may be a plurality of valve members each associated with a respective port, whilst the valved ports are preferably in a "diaphragm" member disposed within the rotor so as to separate the ports in the latter from the atmosphere or from the aforesaid duct leading to the air inlet end of the compressor. When there are several valve members they may operate synchronously gradually to vary the opening of the ports similarly with the speed variation, or they may be arranged to open the ports consecutively, in which case the valve members may operate with snap action.

The valve members may operate in sliding contact with the "diaphragm" member, but preferably they operate by moving to-and-away therefrom. In the former case the diaphragm member may be a radial annulus with axial ports, and in the latter case it may be a radially ported cylindrical sleeve mounted coaxially within the compressor rotor; in either case the ported diaphragm is preferably rigidly connected to or is even integral with the compressor rotor.

In carrying out the invention a ported diaphragm is provided within the rotor, and the diaphragm has associated with it a valve member or valve members by which the extent of opening of the diaphragm ports is varied under centrifugal action.

In one arrangement the diaphragm is a radial or conical or coaxially cylindrical member, and the valve may be a similar washer-shaped or conical member, also having ports, and the relative movements between the ported diaphragm and the valve may be controlled by spring-biased centrifugal members, or the valve or valves may be constituted by one or more springs or by spring-biased blades, all as will hereinafter appear.

In the accompanying drawings:

Fig. 1 is a sectional elevation in a conventional form of the upper half of the high pressure end or section of an air compressor having one form of the invention associated with it,

Fig. 2 is an end view, partly in section, of the arrangement shown in Fig. 1,

Fig. 3 is a view similar to Fig. 1 showing a modified arrangement in accordance with the invention,

Fig. 4 is a section of the cylindrical diaphragm member, taken on the line IV—IV of Fig. 3,

Fig. 5 is a sectional elevation of another modification of a valved diaphragm according to the invention,

Fig. 6 is a sectional end view of the arrangement shown in Fig. 5.

Fig. 7 is a sectional elevation, and Fig. 8 an end view of a still further modification in accordance with the invention.

Referring first to Figs. 1 and 2, at 1 is shown the outer casing of the low pressure stage or section of a compressor carrying the alternate stages of blading as shown and 2 is the rotor carrying the intercalated blading stages as shown. It is to be understood that the casing member 1 may

be arranged to rotate in the opposite direction to the rotor 2 in a contra-rotational compressor.

At 4 and 5 are shown stationary wall members constituting the annular air inlet to the illustrated compressor or compressor section. Integral with, or attached to, the annular member 5 is a diaphragm 5a carrying or integral with a bearing housing or support member 5b (shown purely diagrammatically) carrying bearings 6 for a hollow sleeve shaft 7 which flares at 7a into a cylindrical portion 7b attached to or integral with which is a conical portion 7c having a peripheral flange 7d which is bolted to a flange 2a integral with or secured to the rotor cylinder 2 and provided with a plurality of circularly distributed ports 8. The ported flange 2a constitutes the diaphragm hereinbefore referred to and bolted to it is the internal flange of a conical member 9 integral with or secured to the rotor, a part of which is indicated at 10, of the high pressure compressor. The right-hand end of the rotor cylinder 10 and the left-hand end of the rotor cylinder 2 are shaped and spaced apart as shown to leave the sloping annular duct or a plurality of holes 11 through which can flow the air to be bypassed.

The sleeve portion 7b, 7b has housed within it on ball bearings 12 and 13 an inner sleeve or cylinder 14 in the space between which and the sleeve portion 7b, 7b is a helical spring 15, preferably formed of square section wire, having its right-hand end anchored at 1e to the sleeve 7b, whilst the other end of the spring is anchored at 14a to the sleeve 14. To the left-hand end 14b of the sleeve 14 is secured by one or more keys 16 and nut 17 the hub of a toothed wheel 18.

Meshing with the toothed wheel 18 are two or more unbalanced centrifugal members 19, symmetrically distributed around the rotor flange 2a so that the dynamic balance of the rotor as a whole is not disturbed. Each member 19 is pivoted on a pin 20 with a bearing sleeve 21, the pins 20 being secured by nuts as shown to the diaphragm flange 2a and the members 19 being accommodated in the main in a recess 2b in the flange 2a and covered by the flange of the member 9.

Each centrifugal member 19 has two toothed rim portions 19a and 19b, the teeth of the latter engaging with the wheel 18. The member 19, as clearly shown in Fig. 2 has a cut-away portion on one side and the solid or weighted portion 19c on the opposite side of the axis. It is further provided with the projection 19d which can oscillate between abutments 22 provided either by cutting away a portion of the diaphragm flange 2a or by forming a recess in the flange of the member 9.

The ported valve member is shown at 23 and it is of washer shape, having round the whole (or, as shown, portions) of its periphery teeth 23a which are in permanent mesh with the teeth 19a of the centrifugal member 19. The valve member 23 is at its toothed hub portion 23b of greater thickness, bearing in the periphery of the recess 2b of the diaphragm flange 2a.

As hereinbefore indicated the ports in the valve member 23 are caused by means of the spring 15 to register with the ports 8 in the diaphragm 2a so that when the rotational speed of the plant is less than the predetermined minimum, air will be bypassed through the annular duct 11, through the ports 8 into an annular space 24 provided by the inner surface of the right-hand part of the rotor 2 and by a flaring

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wall 25, the left-hand end of which has a flange 25a between the diaphragm flange 2a and the flange 7d of the shaft 7. The right-hand end of the annular wall 25 is curved as shown at 25b and provided with guide blading 25c, so that the bypassed air is constrained to flow back into the inlet of the low pressure section of the compressor through the gap 26 provided between the stationary wall member 5 and the right-hand end of the compressor rotor cylinder 2. The right-hand end 25b of the wall member 25 is shaped so that the bypassed air may provide an injector effect upon the intaken air whereby to prevent or minimise the stalling effect, namely by appropriately directing bypassed air into the main air flow in the low pressure stage or section of the compressor. Furthermore, the guide blades 25c may be arranged to impart to the bypassed air a relatively circumferential component of motion suited to the angle of the blading of the first row of rotating compressor blades. In addition, or alternatively, the guide blading 25c or boundary members may direct the flow of the bypassed air slightly outwardly at such angle that there will be created an annular zone or dead space, as indicated by the dotted line 27, at the radially inner ends of the compressor blades, which are to some extent inactive with respect to the main air flow, whereby to give in effect a shortening of the blade heights so as to reduce the effective annular blade area presented to the main air intake and to the passage of this main air through at least the earlier rows or stages of blading.

Referring next to Figs. 3 and 4 of the drawings, it will be seen that, apart from minor differences, the arrangement therein illustrated differs in the main from that illustrated in Figs. 1 and 2 in that the ported diaphragm 2a of these earlier figures, instead of being a radial flange, is a cylindrical member 28, having arcuate ports 28a, of which about any one radial plane there are four in number as clearly shown in Fig. 4, whilst there are four sets of such ports distributed axially along the cylinder 28 as clearly shown in Figure 3.

The cylinder 28 is constructed separately from the rotor 2', and is connected to the latter by means of screws 29 passing through the flange 28b to a conical annular member 30, the right-hand end of which is secured by screws 31 to the conical disc 32, which, at its flanged outer periphery, is screwed to an annular lug 2c on the rotor 2'.

The conical disc 32 forms part of the flaring end 32a of the shaft 32b of the rotor 2' and is provided with ports 32c through which air which is permitted to flow through the ports 28a of the cylindrical diaphragm 28 can pass as constrained by the wall member 25d, the outer periphery of which extends into the gap 26 between the stationary member 5 and the right-hand end of the rotor 2' in a manner which may be similar to that shown in and described with reference to Fig. 1. The inner periphery of the wall member 25d is secured to the flaring portion 32a of the rotor shaft 32b.

In Fig. 3 the rotor 2' is shown as formed at the left-hand end of the low pressure part or section with an annular slit or a plurality of holes 2d through which at the gap 11, the air to be bypassed can flow, thence leaving through the holes 2e into the space within the compressor rotor 2' wherein it is confined and is caused to flow in the direction indicated generally by the

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long arrow, by a transverse wall member 25e the outer periphery of which may be conical and is secured to the inner surface of the rotor 2'.

At 33 is shown another ported cylinder fixed within the diaphragm cylinder 28 by a flange 33a and the screws 29. The ports 33b of the inner cylinder 33 are relatively radially staggered to the ports 28a in the diaphragm cylinder 28.

The diaphragm cylinder 28 is formed interiorly with a plurality of radial lugs 28c which serve as guides for the valve members 34, of which there are four in number as clearly shown in Fig. 3. Each valve member 34 is in the form of a C-spring and co-operates with one peripheral row of the ports 28a in the diaphragm member 28. Each spring valve 34 preferably subtends the most part of 360°, its adjacent ends, when in its contracted condition, meeting the opposite sides of an abutment 35 riveted to the inner ported cylinder 33 (or may be to the diaphragm cylinder 28), the springs, when in contracted condition, more or less closely embracing the inner cylinder 33, all as clearly shown in Figs. 3 and 4. The bypassed air from the holes 2d and 2e flows radially through the ports 33b in the inner cylinder 33, then taking axial component of direction to flow substantially radially out through the ports 28a in the diaphragm 28.

It will be understood that as the rotational speed of the plant increases the normally open ports 28a will be more and more closed and finally completely closed at the predetermined speed by reason of the springs 34 expanding under centrifugal force.

In order to preserve dynamic balance the spring valves 34, if two or more in number, have the gaps therein evenly circularly distributed in the axial direction along the diaphragm cylinder 28.

In order that each spring 34 will, in its fully extended condition, conform to the inner surface of the diaphragm cylinder 28, the radial thickness of the spring 34 may vary, being thickest at the centre of its peripheral length, as will be well understood.

In the arrangement illustrated in Figs. 5 and 6 of the accompanying drawings, which arrangement is a modification of that illustrated in Figs. 3 and 4, the diaphragm cylinder 28 is shown as of shorter axial length and provided with one circular row of four ports 28a, each having co-operating with it a centrifugal (unbalanced) valve member comprising a substantially rigid arcuate blade 36 integral with a shaped hub portion 36a, having a bearing sleeve 36b and carried on an axle 37, the end of which is engaged in a perforation in two washer-shaped members 38 and 39 which have axially extending spacer portions 38a and 39a and which are secured together by a plurality of bolts 40. The bolts 40 secure the members 38 and 39 to an internal flange 28d of the ported diaphragm member 28. The axes 37 are carried in lugs 38b extending radially from the members 38 and 39.

Also housed between the members 38 and 39 is an annular rotatable member 41 which has for each valve a radial bifurcation 41a engaging a pin 42 fixed to a radially inwardly extending portion 36c of the hub portion of each valve member 36, of which the blade portion constitutes the centrifugal mass, being thrown outward at the predetermined speed to lie flush against the inner surface of the diaphragm cylinder 28, thereby closing the ports 28a therein.

It will be appreciated that instead of the bifurcation 41a the relatively rotatable member 41

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may be provided with teeth engaging corresponding teeth on the valve hub 36a in substitution for the pin 42.

The relatively rotatable member 41 is biased by means of a pair of C-springs 43 and 44, one end of each of which is anchored as at 43a to the member 41, whilst the other end is anchored as indicated at 44a to the member 39: the arrangement is such that balance is obtained during rotation. It will be appreciated that the springs 43 and 44 inherently contract themselves inwards and expand under centrifugal action. Alternatively, however, springs may be used which normally tend to expand outwards, in which case each spring, as shown in Figs. 7, 8 at 43b, 44b, has one of its ends in contact with a fixed abutment 45 and its other end against an abutment 41b on the relatively rotatable member 41c, which is shown as having teeth 41d, which, as described as an alternative with reference to Figs. 5 and 6, engage corresponding teeth on the portions 36c of the hub portions 36a of the valve members 36. Otherwise the arrangement is the same as that illustrated in Figs. 5 and 6, whilst it will be appreciated that a separate spring may be provided for biasing each of the valves.

I claim:

1. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, and means operating automatically to effect the opening of said last named means when said casing is rotating at a speed predeterminately less than the optimum speed of the compressor.

2. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, said last named means including a member within said inner casing and having at least one valve port, and valve means operative to effect the opening of said port when said casing is rotating at a speed predeterminately less than the optimum speed of the compressor.

3. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, said last named means including a member within said casing and having at least one normally open valve port, and valve means for gradually closing said port as the speed of said casing increases above a predetermined minimum speed and fully closing said port at a predetermined higher speed.

4. A plant according to claim 2 wherein said valve means is biased to open position.

5. A plant according to claim 2 wherein said valve means is normally open below a predetermined speed of said casing and are provided with means operated by centrifugal force for closing said valve means.

6. A plant according to claim 2 wherein said valve port is in a member rotatable with said casing.

7. A plant according to claim 2 wherein said valve port is in a member rotatable with said casing and said valve means is also rotatable with said casing.

8. A plant according to claim 2 wherein said valve means is normally urged to open position by spring means.

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9. A plant according to claim 2 wherein said valve means is normally urged to open position by spring means and unbalanced centrifugal means acts in opposition to said spring means.

10. A plant according to claim 2 wherein said valve means is constituted by a spring.

11. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, means operating automatically to effect the opening of said last named means when said casing is rotating at a speed predeterminately less than the optimum speed of the compressor, and means for returning air withdrawn from said intermediate stage to the inlet end of the compressor.

12. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, means operating automatically to effect the opening of said last named means when said casing is rotating at a speed predeterminately less than the optimum speed of the compressor, and means for returning air withdrawn from said intermediate stage to the inlet end of the compressor, said last named means including means whereby the returned air will provide an injector action at the inlet end of the compressor.

13. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, means operating automatically to effect the opening of said last named means when said casing is rotating at a speed predeterminately less than the optimum speed of the compressor, and means for returning air withdrawn from said intermediate stage to the inlet end of the compressor, said last named means including guide blades for imparting a circumferential component of motion to the air returned to the inlet end of the compressor.

14. A gas turbine plant including a multi-stage axial flow compressor having an inner rotating casing, means associated with said inner casing through which air may be withdrawn from an intermediate stage of the compressor, means operating automatically to effect the opening of said last named means when said casing is rotating at a speed predeterminately less than the optimum speed of the compressor, and means for returning air withdrawn from said intermediate stage to the inlet end of the compressor, said last named means including means for directing the air returned to the inlet end of the compressor slightly away from the inner wall of the compressor.

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