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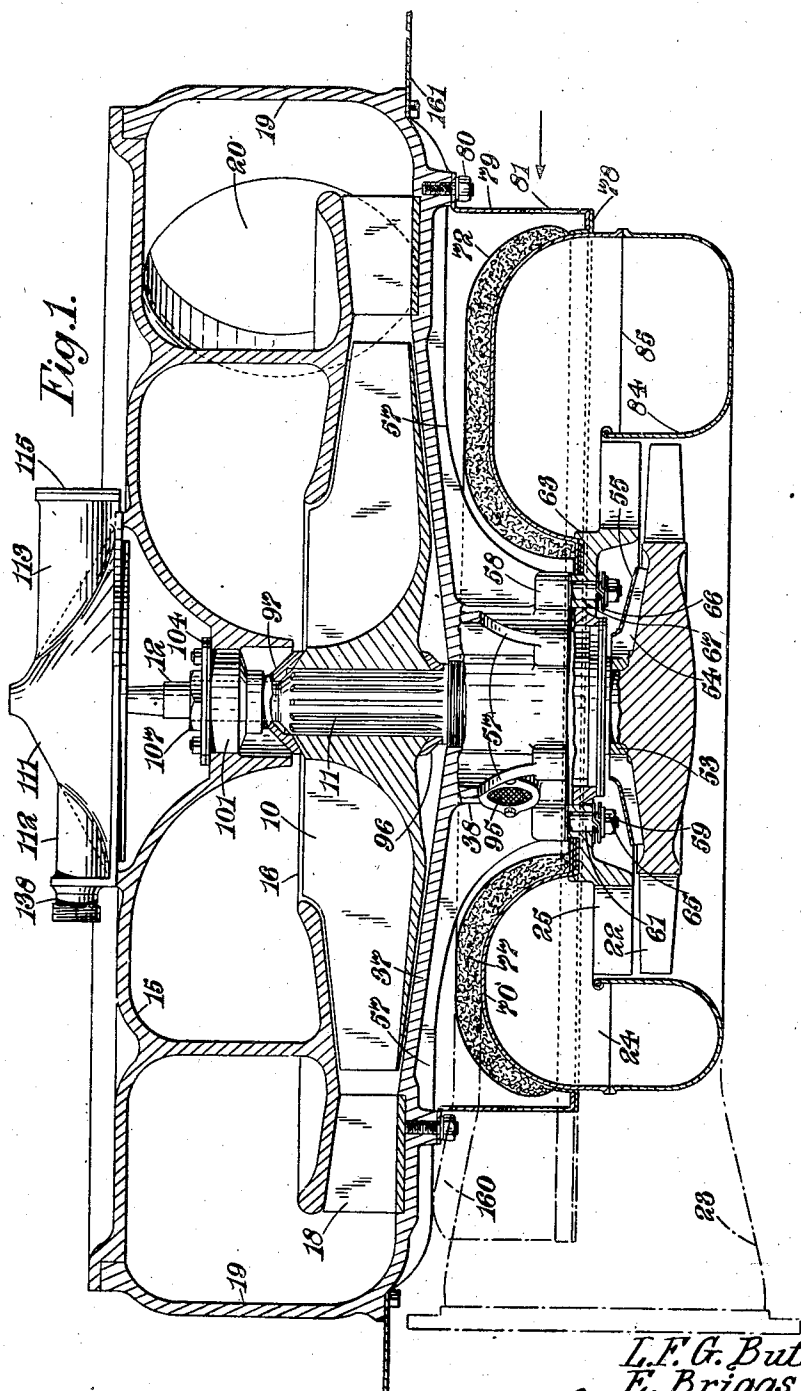
L. F. G. BUTLER ET AL

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GAS TURBINE

Filed Feb. 20, 1940

6 Sheets-Sheet 1



Inventors
L. F. G. Butler
E. Briggs
and E. Bowness
by Wilkinson & Mawkinney
Attorneys.

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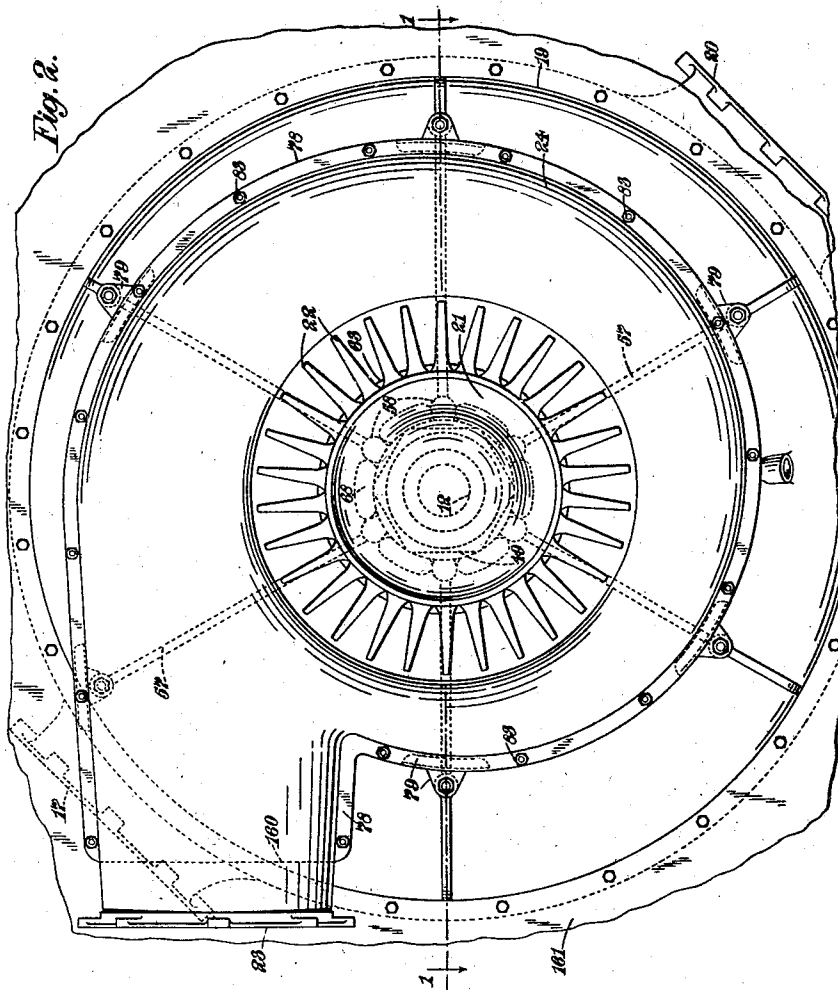
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6 Sheets-Sheet 2



Inventors
L. F. G. Butler
E. Briggs
and E. Bowness
by Wilkinson & Mawhinney
Attorneys.

Sept. 22, 1942.

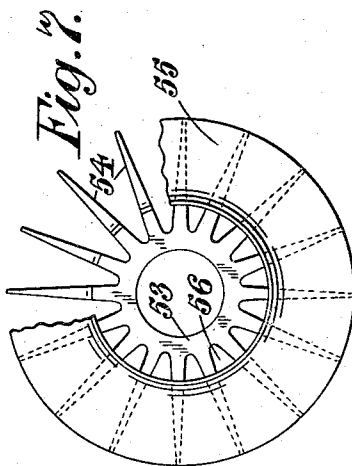
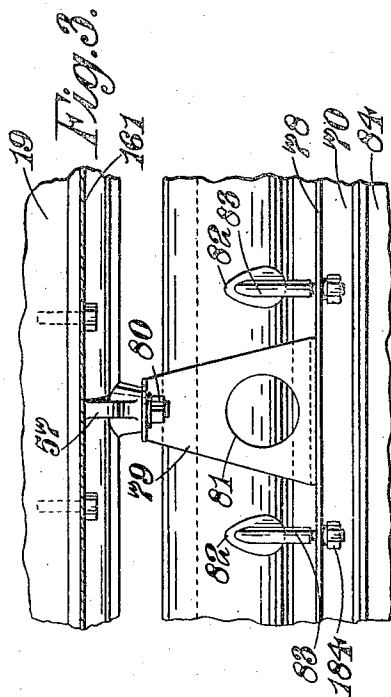
L. F. G. BUTLER ET AL

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Inventors
L. F. G. Butler
E. Briggs
and E. Bowness
by Wilkinson & Mawhinney
Attorneys.

Sept. 22, 1942.

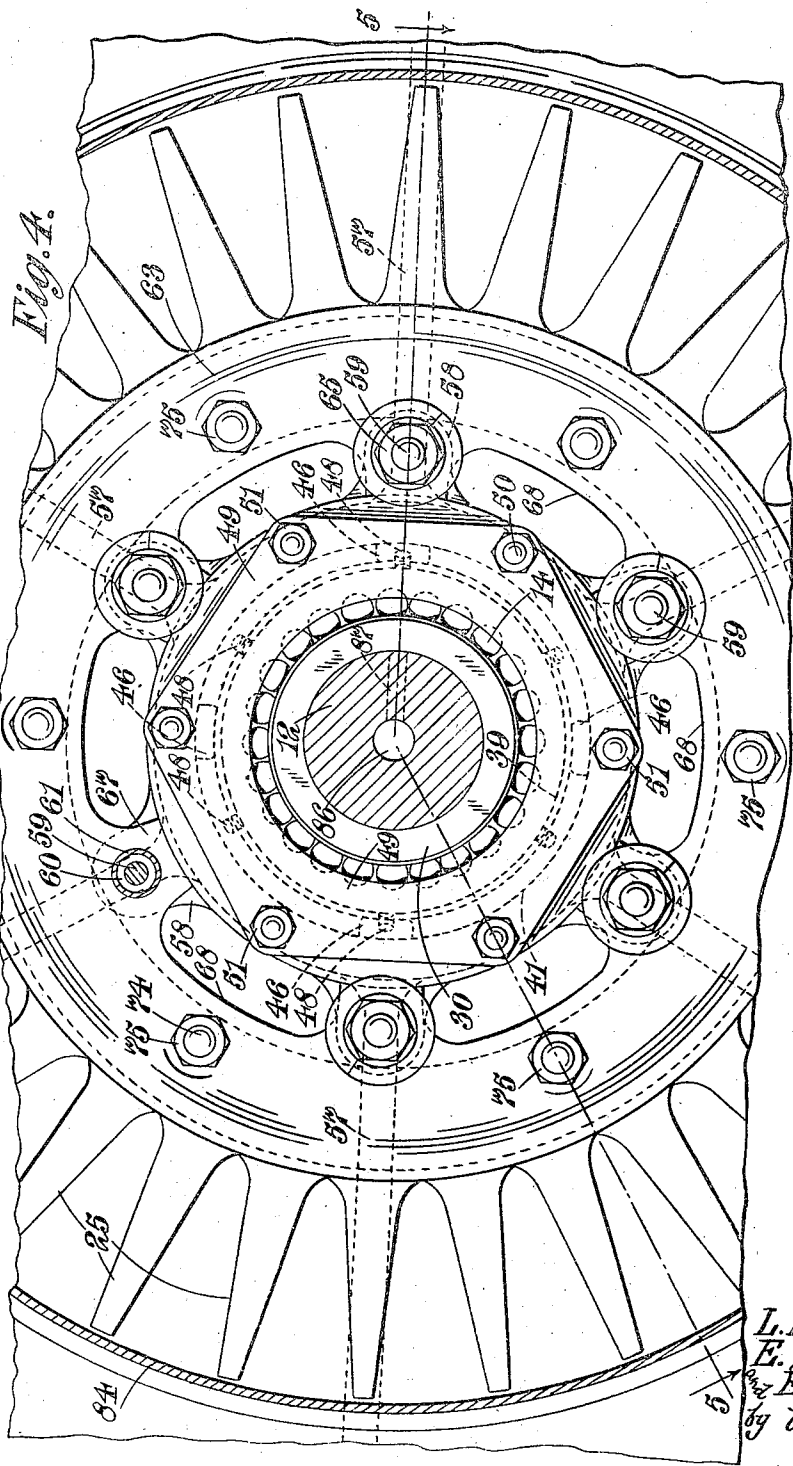
L. F. G. BUTLER ET AL

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Inventors
L. F. G. Butler
E. Briggs
E. Bowness
by Wilkinson &
Mawhinney
Attorneys.

Sept. 22, 1942.

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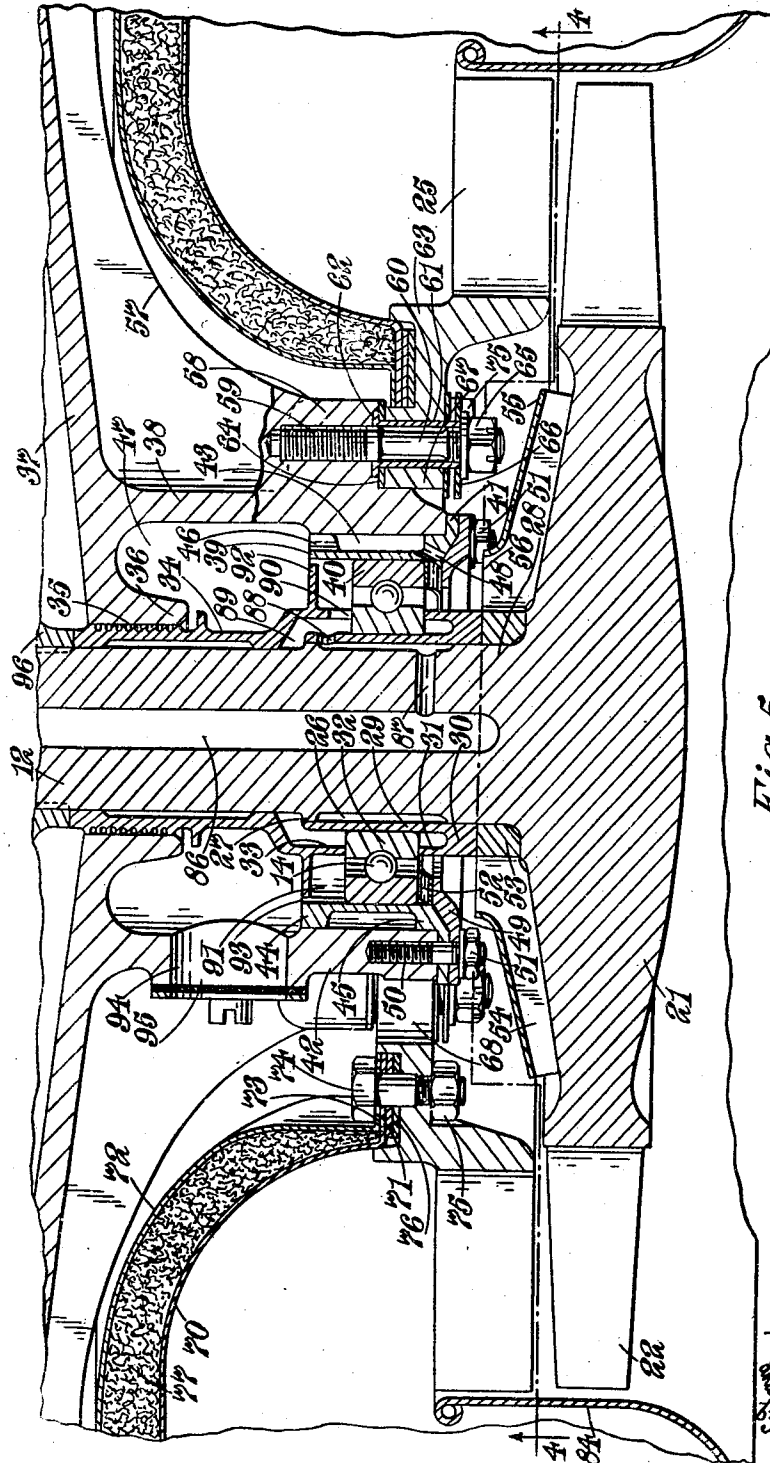


Fig. 5.

Inventors
L. F. G. Butler
E. Briggs
E. Bowness
by Wilkinson &
Mawhinney
Attorneys.

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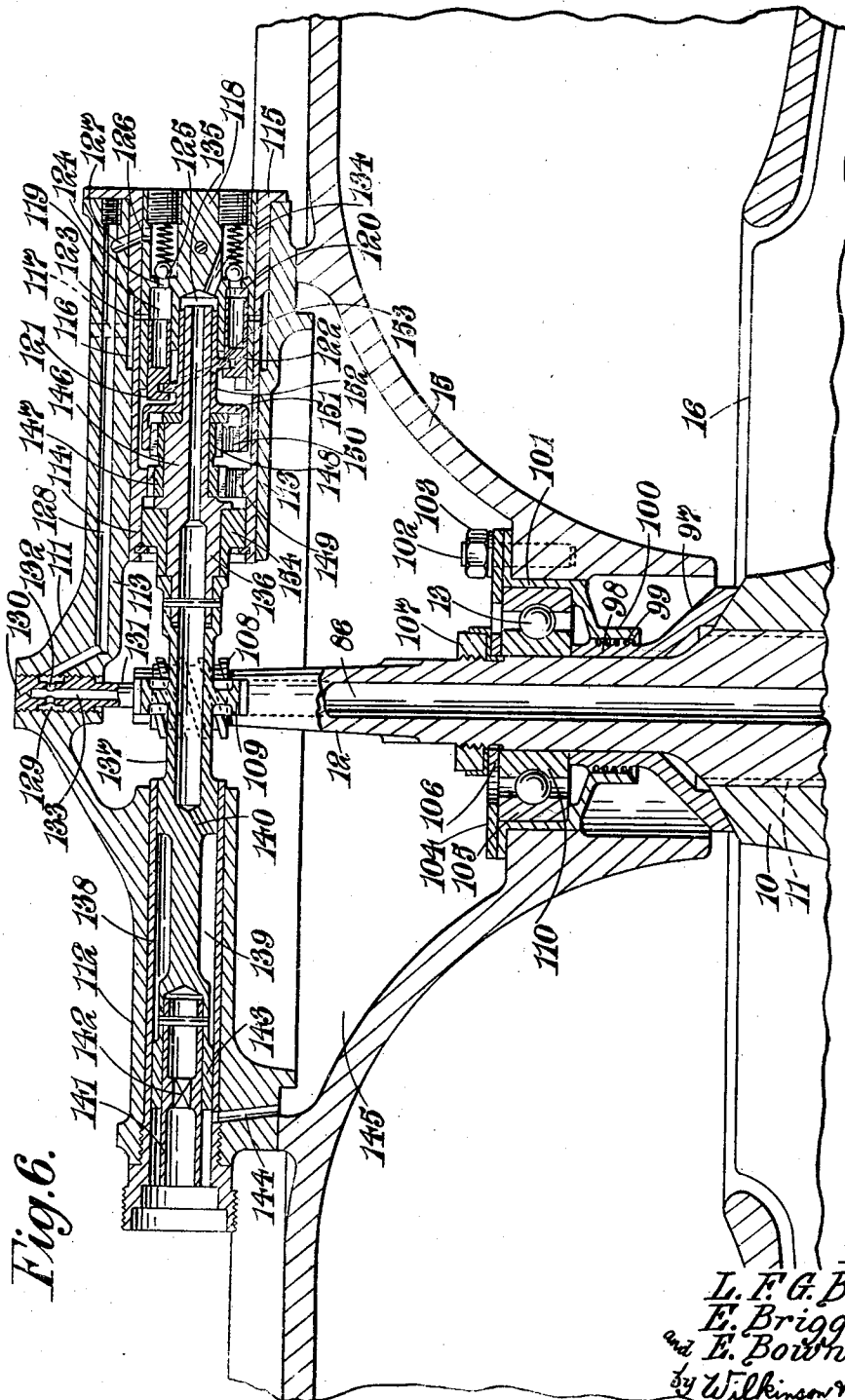
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Inventors
L. F. G. Butler
E. Briggs
and E. Bowness
by Wilkinson & Mawhinney
Attorneys

UNITED STATES PATENT OFFICE

2,296,701

GAS TURBINE

Leonard Frederick George Butler, Ernest Briggs,
and Edward Bowness, Bristol, England, assign-
ors to The Bristol Aeroplane Company Limited,
Bristol, England, a British company

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4 Claims. (Cl. 230—116)

This invention relates to gas-turbines such as may be used for driving the supercharging compressor of an internal-combustion engine, the turbine being driven by the exhaust gases of the engine.

The object of the invention is to provide improved cooling and lubrication of the working parts.

According to the invention, a gas-turbine comprises an anti-friction bearing disposed closely adjacent the rotor and means for circulating cooling fluid, such as air, through a space between the outer race of the bearing and the surrounding structure and/or through a space between the inner race of the bearing and the rotor shaft. The said flow of fluid is preferably induced by means of a centrifugal fan the blades of which are mounted adjacent the rotor, on the bearing side thereof, so as to draw air through the bearing and discharge it over the adjacent face of the turbine rotor. It is already known to provide a centrifugal fan between a turbine rotor and its supporting bearing but there is a tendency for such a fan to produce mere turbulence unless its points of intake and discharge are arranged in definite relationship to the bearing and the rotor blades respectively. According to another feature of the invention, the blades of the fan are shrouded on both sides, the blades being shrouded on the turbine rotor side by abutting against the adjacent face of the rotor. The gas-turbine is of the axial-flow type so that air is compelled to flow into the central eye of the fan and out through the inter-blade spaces.

The gases for driving the turbine are supplied to an annular nozzle, having a ring of fixed guide-vanes, by a scroll-shaped chamber. Where this chamber is disposed on the same side of the rotor as the bearing aforesaid there is a tendency for the heat of the gases to flow to the bearing. Means may be provided, according to the invention, for reducing this flow of heat.

Where the turbine drives a supercharging compressor for an internal-combustion engine, the lubrication requirements of the turbine bearing have no fixed relation to the engine speed. A turbo-compressor for supercharging an internal-combustion engine, according to another feature of the invention, therefore comprises one or more lubricating pumps driven by the turbine shaft so as to supply a metered feed of lubricant to the bearing. Where the compressor comprises a centrifugal fan mounted on the same shaft as the turbine rotor the common shaft may be supported in two or more bearings spaced apart

lengthwise of it and there may be a separate pump or group of pumps appropriated to each bearing.

The space, above referred to, between the shaft and the inner race of the turbine bearing may be supplied with cooling liquid from one of the said pumps; the said liquid is preferably lubricating oil which also lubricates the said bearing.

A specific embodiment of the invention will now be described, by way of example, with reference to the accompanying drawings in which:

Figure 1 is a sectional elevation of a turbo-compressor for use in conjunction with a supercharged aircraft engine, Figure 1 being a section on the line 1—1 of Figure 2,

Figure 2 is an underplan of the installation shown in Figure 1,

Figure 3 is an elevation of a detail, being a view in the direction of the arrow 3 of Figure 1,

Figure 4 is a view similar to Figure 2, drawn to an enlarged scale, being a section on the line 4—4 of Figure 5,

Figure 5 is a sectional elevation corresponding to Figure 4 showing the turbine bearing and associated parts, being a section on the line 5—5 of Figure 4,

Figure 6 is a sectional elevation, also drawn to the same scale as Figure 4, showing the lubricating pump, the impellor bearing and associated parts, and

Figure 7 is a plan, to the same scale as Figure 1, showing the fan which is carried by the turbine rotor.

The general arrangement of the turbo-compressor will now be described with reference to Figures 1, 2 and 3. The impellor 10 of a centrifugal blower of known or usual construction is attached by splines 11 and other means (described below) to a shaft 12 which rotates in a top bearing 13 (Figure 6) and a bottom bearing 14 (Figure 5). The top bearing is supported in the volute chamber 15 through which air, or a mixture of fuel and air, is admitted to the eye 16 of the impellor. The air enters the chamber through an air-inlet 17 (Figure 2) and is discharged by the impellor 10 through fixed diffuser vanes 18 into a collecting chamber 19 from which it flows through an outlet 20 to the induction system of the internal-combustion engine. The compressed air may either be discharged directly to the induction system or, more usually, it will flow to the inlet of a carburettor from which a mixture of fuel and air is admitted to a second-stage blower which discharges the compressed mixture to the engine cylinders. The compressor

of the present specification is therefore preferably a first-stage compressor in a two-stage system, but the invention is not limited to this arrangement.

The bottom of the shaft 12 is formed integrally with a turbine rotor 21 having blades 22 through which the exhaust gases from the internal-combustion engine are made to flow, being first admitted through a pipe 23 to a scroll-shaped chamber 24 from which they are discharged downwardly through fixed guide-vanes 25 to the blades 22. The gases discharged from below the rotor blades 22 may be suitably guided by an exhaust-pipe but, where the invention is applied to the turbo-compressor for an aircraft engine, as above stated, the gases may simply be discharged into the external airstream flowing along the underside of the aircraft fuselage or wing.

The detailed construction of the turbine bearing 14, and associated parts, will now be described with reference to Figures 4 and 5. The shaft 12 is cut with a groove 26 surrounding it near the rotor 21, the groove being bounded at its upper end by a shallow flange 27 on the shaft and at its lower end by an unreduced portion 28 lying adjacent the rotor disc. The groove 26 is surrounded by a thin steel sleeve 29 the bottom end of which, however, is thickened at 30 and has integrally formed with it a short vertical cylindrical portion 31 the top of which engages the bottom of the inner race 32 of the ball-bearing 14. Engaging the top of the race 32 is the downwardly-extending tubular part 33 of a sleeve 34 surrounding the shaft. Near its upper end the sleeve 34 is cut with a series of grooves 35 which form a packing between the sleeve (which rotates with the shaft) and a surrounding boss 36 which is formed integrally with the bottom wall 37 of the compressor casing 19. The wall 37 also has formed integrally with it a downwardly-extending tubular casing 38 which supports the outer race of the turbine bearing and the fixed guide-vanes 25 as will now be described. Surrounding the outer race 40 of the bearing 14 is a steel sleeve 39 of which the bottom end is flanged outwardly as shown at 41 to abut the underside of the bottom portion 42 of the tubular casing 38. The sleeve 39 is cut with a groove 43 extending right round it so as to leave shoulders 44, 45 at the top and bottom respectively each engaging the inner wall of the part 42. As shown in Figure 4 the shoulder 44 is cut with four notches 46 spaced equally around it so as to provide communication between an air-space 47 inside the tubular casing, and the groove 43 referred to. The shoulder 45 is cut with six oblique holes 48, equally spaced around the sleeve, each leading from the groove 43 to a space below the outer race 40 of the bearing. The sleeve 39 is held in place by an annular plate 49 fastened by studs 50 and nuts 51 to the bottom of the part 42, the studs 50 also passing through the flange 41 on the bottom of the sleeve 39. The inner edge of the plate 49 underlies the outer race 40 of the bearing which is separated from it by a waved springy washer 52. The bearing is thus supported by the plate 49.

The thick portion 30 at the bottom of the sleeve 29 is separated from the disc 21 of the turbine rotor by the hub 53 of a centrifugal fan having blades 54 (see also Figure 7) the lower edges of which abut against the top face of the rotor disc and the upper edges of which are shrouded by a conical shroud member 55 so as to form a central annular eye 56 lying beneath 75

the bearing 14 and a series of radial tunnels, each bounded at the bottom by the rotor disc 21, at the top by the shroud 55 and at each side by two adjacent blades 54.

The bottom plate 37 of the compressor casing is formed with six symmetrically-disposed reinforced ribs 57, at the innermost and lowermost point of each of which a boss 58 is formed to receive the screw-threaded end of a stud 59. The stud is formed with a reduced portion 60 in its length which passes through a sleeve 61 having a flange 62 at its upper end, the flange 62 being countersunk into the bottom face of the boss 58. The sleeve 61 engages a hole in one of a series of arms or fingers 67 extending inwardly from a ring 63 with which the guide-vanes 25 are integrally formed, the top face of each finger 67 being separated from the boss 58 by a washer 64 of asbestos or other heat-insulating material. The lower end of each stud 59 is engaged by a nut 65 which, however, is separated from the finger 67 by a spring-washer 66. It will be seen from Figure 4 that the fingers 67 are separated from one another by wide scallops or recesses 68 the purpose of which is explained below. In Figure 4 one of the studs 59 and its associated sleeve 61 are shown in section so that the shape of the finger 67 in relation to the boss 58 can be clearly seen.

The scroll-shaped chamber 24 to which the exhaust gases are admitted is supported from the ring 63 in the following manner. The top inside wall 70 of the chamber is formed with a flange 71 at its inner edge and an outer top wall 72 is similarly flanged at 73 the flanges 71 and 73 being engaged by a series of bolts 74 which pass through the ring 63 and are engaged by nuts 75. The flange 71 is, however, separated from the base of the recess in the ring 63 in which the flanges lie by two or more gaskets 76 of asbestos or other heat-insulating material. The two top walls 70 and 72 of the exhaust chamber are spaced apart and the space between them is occupied by a lagging 77 of suitable non-conducting material.

The inside upper wall 70, at its outer edge, carries a ring 78 welded to it (see also Figure 3); welded to the ring 78 is a plurality of brackets 79 of which the upper ends are turned over and fastened to the bottom plate 37 of the compressor casing by the nuts 80 of suitable studs. Each bracket 79 is perforated as shown at 81 and is of such lightness and flexibility as will permit relative movement between the exhaust-gas chamber and the compressor casing in the manner described below. As shown in Figure 3 small plates 82 are welded to the outer wall 72 of the scroll-shaped chamber; each of these plates carries a bolt 83 passing through the flange 78 and engaging a nut 84. In this manner the two upper walls 72 and 70 of the exhaust chamber are secured together by their outer periphery.

The bottom wall 84 of the exhaust chamber is supported by being welded along the line 85 to the inner upper wall 70.

It will be explained below how a metered flow of lubricating oil is supplied along an internal conduit 86 in the shaft 12, through a radial passage 82, to the groove 26. From this groove the oil passes through a port 88 in the steel sleeve 29 into a space 89 above the inner race 32 of the bearing. From the space 89 the oil flows through one or more ports 90 in the depending cylindrical portion 33 of the sleeve 34 into the space 91 immediately above the bearing. The oil flows

down from this space through the bearing, through the central hole in the plate 49.

The sleeve 34 is formed with a flange 92 of slightly less external diameter than the internal diameter of the sleeve 39 so as to leave a narrow annular space 93. Air from the chamber 47 can flow through this space into the space 91. Air enters the chamber 47 through a lateral port 94 to which a suitable air-filter 95 is fitted.

As shown in Figure 1 the impellor 10 of the compressor is engaged on its lower side by a cone-shaped plate 96 and on its upper side by a similar plate 97 (see Figure 6). The plate 97 is formed with a sleeve-shaped portion 98 which is cut with a series of grooves 99 to form packing between the sleeve and a downward extension 100 from a ring 101 which is attached to the wall 15 of the inlet volute by studs and nuts 102, 103. The nuts also locate an annular plate 104 between which and a shoulder at the bottom of the ring 101 the outer race 105 of the impellor bearing 13 is clamped. The inner race is engaged at the bottom by the top of the sleeve 98 and at the top by a washer 106 which is held in place by a nut 107 screw-threaded to the shaft 12. At its upper end the shaft 12 carries a worm 108 which meshes with a worm-wheel 109 whereby the oil pumps (described below) are driven.

The turbo-compressor shaft is assembled in the following manner. The turbine bearing 14 having been assembled in the surrounding casing, and the sleeve 34 having been inserted through the boss 36, the fan-hub 53 and the part 29, 30 are slid on to the shaft 12 from its upper end and the shaft is then introduced through the turbine bearing until it occupies the position shown in the drawings. The cone-shaped plate 96 is then slid on, then the impellor 10, then the plate 97. The impellor bearing 13 is now assembled into the position shown in Figure 6 and the washer 106 is slid on to engage the inner race 110. The nut 107 is now tightened up whereby the race 110, the sleeve 98 and plate 97, the impellor hub, the plate 96, the sleeve 34, the race 32, the parts 31 and 30, and the fan-hub 53 are all clamped between the nut 107 and the turbine rotor disc 21.

The oil pumps will now be described. Engaging the top of the impellor casing 15, as shown in Figure 1, is a conical closure plate 111 which is formed at one side with a tubular enlargement 112 and at the other side with an enlargement 113 in which the pumps are accommodated. The enlargement 113 is engaged internally by a sleeve 114 of which the outer end is flanged at 115, the flange being affixed to the plate 111 by suitable bolts (not shown). Between the sleeve 114 and the surrounding wall of the part 113 an annular space 116 is formed into which a conduit 117 leads from a reservoir (not shown) of lubricating oil. The end of the sleeve 114 is engaged by a cylinder-block 118 which is cut with two cylindrical recesses 119, 120 to accommodate pistons 121, 122, respectively. The cylinder 119 and the surrounding sleeve 114 are cut with a conduit 123 to form an inlet port from the groove 116 to the interior of the cylinder. This port is controlled by the piston 121 in a manner described below. The delivery port 124 of the cylinder 119 is controlled by a spring-pressed ball-valve 125 of which the valve-chest communicates through a port 126 and a conduit 127 in the sleeve 114 and casing 113 with a conduit 128 in the casing. This conduit leads to a groove 129 in a plug 130 which closes the top of the conical casing 11 and

with which a stem 131 is integrally formed and leads down into the interior of the shaft 12. Radial ports 132 connect the groove 129 with the bore 133 of the stem. Thus the delivery of the pump-cylinder 119 is supplied to the interior of the shaft 12. The other cylinder is equipped with a similar delivery port and ball-valve the valve-chest of which is connected by a conduit 134 to a central space 135 in the cylinder block 118. From this space the oil is delivered along the interior of a shaft 136 which is cross-pinned to the shaft 137 with which the worm-wheel 109 is integrally formed. The shaft 137 where it enters the sleeve 138 which lines the enlargement 112 is formed with a groove 139 which is connected by a port 140 with the interior of the right-hand end of the shaft. At its left-hand end the shaft 137 is cross-pinned to a shaft 141 having a non-circular portion 142 for engagement with the end of a flexible shaft from which a tachometer is driven to indicate the rate of revolution of the turbo-compressor shaft. Oil from the space 139 lubricates the bearing between an enlarged portion 143 of the shaft 137 and the surrounding sleeve 138 and, having passed beyond this portion, escapes through a conduit 144 in the casing 111 into the cup-shaped chamber 145 formed by the central portion of the compressor casing 15.

The shaft 137 is formed with an eccentric portion 146 which constitutes an axle for a compound planetary pinion comprising gears 147 and 148. The gear 147 meshes with a fixed internal gear 149 cut on the inside wall of the sleeve 114. The gear 148 meshes with an internal gear 150 cut on a cup-shaped member 151 carried by a short shaft 152. The shaft 152 is formed integrally with a face cam 153 which engages simultaneously with notches in the pistons 121, 122 whereby, upon rotation of the cam, the pistons are reciprocated in their cylinders. The shaft 152 is borne in the central recess 135 in the cylinder block and the shaft 136 is borne at its right-hand end in the shaft 152 and at its left-hand end in a bush 154 in the inner end of the sleeve 114.

As the piston 121, for example, is moved to the left as seen in Figure 6 the ball 125 closes and a vacuum is produced in the cylinder 119 until the port 123 is uncovered. Thereupon oil from the space 116 flows into the cylinder and on the succeeding stroke to the right this oil is supplied through the delivery port 124 in the manner already explained. The pump 122 operates in the same manner.

It will be understood that the shaft 12 rotates at a high speed and the gearing 108, 109 and the epicyclic gearing 146, 147, 148, 149, 150 ensures that the pump-plungers 121, 122 are driven at a suitably reduced speed.

The oil which enters the space 145 through the port 144 flows downwardly through the bearing 13 and lubricates the balls escapes slowly through the packing 99, 100, and is discharged by centrifugal force from the bottom edge of the cone-shaped plate 97. At this point the oil enters the stream of air flowing into the eye 16 of the impellor.

The manner in which the various features described above with reference to the drawing contribute to the cooling of the turbine bearing will now be described. It will be understood that the fan 54, 55, which is carried by the turbine rotor will suck air downwardly through the central aperture of the plate 49 from between the races 32, 40 of the bearing 14. This will create a suction below the plate 49. Air will therefore flow

from the chamber 47 through the notches 46 in the sleeve 39, into the groove 43. From this groove the air will flow through the holes 48 to the eye of the fan. In passing through the groove 43 the air insulates the outer race 40 of the bearing from heat which would otherwise be conducted to it from the surrounding casing 42. It will be seen that the groove 43 extends in a direction parallel with the axis of the shaft by a distance greater than the dimension of the race 40 in this direction so that the only heat which is conducted from the part 42 of the casing directly to the outer race 40 of the bearing must pass either through the uncut portions of the shoulder 44 or through the shoulder 45 of the sleeve 39.

Another path for air exists through the narrow space 93 into the space 91. From this point the air flows through the bearing in company with the oil and enters the eye of the fan. The suction which exists at the lower side of the plate 49 causes yet a third stream of air to flow between the bottom plate 37 of the compressor casing and the top of the exhaust chamber, radially inwardly and downwardly, through the slot-shaped spaces 68 between the fingers 67, and into the eye of the impellor. It will be seen that the only way in which heat can be conducted from the vane-ring 63 to the part 42 of the bearing casing is along the fingers 67. The area of contact is reduced to a minimum by reducing the diameter of each of the studs 60 where it passes through the sleeve 61 and, where the faces of the parts 67 and 58 abut, an insulating washer 64 is interposed.

The heat can also flow from the blades 22 of the rotor, through the rotor disc 21 to the shaft 12 and from the shaft 12 to the bearing, but conduction of heat along this path is reduced to a minimum as follows.

The oil which is pumped into the conduit 86 within the shaft 12 flows out through the radial passage 87 into the space 26, through the port 88 into the space 89, through the port 90 into the space 91 and then flows through the bearing to lubricate it and is finally discharged into the eye of the fan 54. The oil is ultimately discharged with the exhaust gases into the airstream. The rate of flow of oil is very small in relation to the total consumption of engine oil so that this loss is not serious. The space 26 insulates the race 32 of the bearing from the adjacent part of the shaft 12 and it will be seen that the groove 26 extends in a direction lengthwise of the shaft-axis considerably beyond the bearing on either side. Thus, the only path by which heat can be conducted from the shaft to the inner race of the bearing is through the flange 27 or the unreduced portion 28 of the shaft and then along the sleeve 29. The oil which flows continuously through the groove 26 keeps down the temperature of the sleeve 29.

The differences of temperature to which the installation is subjected give rise to relative movements which are accommodated in the following manner. The expanding and contracting of the exhaust-collecting scroll 24 in relation to the impellor casing are accommodated by the flexible brackets 79 which are sufficiently resilient for this purpose as has already been explained. The shaft 12 is suspended by the impellor bearing 13, the turbine bearing 14 accommodating only lateral thrusts. Any vertical movement of the shaft with respect to the surrounding casing is accommodated by movement of the outer race 40 of the turbine bearing with respect to the

sleeve 39. The spring-washer 52 can expand and contract to accommodate such movements.

The conduction of heat from the exhaust system to the compressor casing is reduced by the lagging 77 already described. In addition, the inward and downward flow of air between the wall 72 of the exhaust chamber and the bottom wall 27 of the compressor casing keeps the compressor casing cool. The lagging referred to must be extended in order to cover the pipe 23 and for this purpose the flange 78 is shaped as shown in Figure 2 so as to conform to the maximum diameter of the pipe 23 and, as shown diagrammatically in Figure 1, the lagging 77 may be extended to form a saddle 160 engaging the upper half of the pipe 23.

The compressor casing may also carry an outwardly-extending plate 161, of any suitable radius, to prevent fuel or other combustible matter from dripping on to the hot exhaust system from the engine above.

The various means, above described, for insulating the turbine bearing from the hot parts of the exhaust system ensure that the oil in the bearing is not disintegrated. In an alternative form of the invention, not shown in the drawings, the space 26 between the inner race and the shaft could be cooled by air instead of by the lubricating oil. The arrangement whereby the lubricating oil pumps are driven by the turbine shaft ensures that a metered flow of oil is supplied to the bearings at a rate depending upon the bearing requirements.

Although the invention has been described in relation to ball-bearings for the turbo-compressor shaft it can be applied to an installation having anti-friction bearings of the cone or roller type.

Moreover, the invention is applicable to turbines and turbo-compressors generally and is not limited to the superchargers of internal-combustion engines.

We claim:

1. A gas-turbine comprising a turbine rotor and an anti-friction bearing disposed closely adjacent the turbine rotor, walls forming a space between the outer race of the bearing and the surrounding structure and a centrifugal fan mounted adjacent the rotor, at the bearing side thereof, so as to draw air through the bearing and through the said space and discharge it over the adjacent face of the turbine rotor.

2. In a turbo-compressor for a supercharged internal-combustion engine, the combination of a shaft, a turbine rotor mounted thereon, an air-impeller also mounted thereon, a supporting structure for said shaft, a sleeve surrounding but spaced away from said shaft adjacent the turbine rotor, a wall secured to but spaced away from said structure opposite said sleeve, an anti-friction bearing having an inner race engaging said sleeve and an outer race engaging said wall, a centrifugal fan mounted adjacent said turbine rotor between it and the anti-friction bearing, means supplying oil to the interior of said sleeve and to said bearing, means admitting air to said bearing and to the exterior of said wall and openings through which the air and oil are extracted by said fan.

3. An axial-flow gas-turbine comprising a rotor, a centrifugal fan driven by the rotor having blades, shrouds on the sides of the blades, an anti-friction bearing for the rotor, one of said shrouds having an eye opening at one side to the bearing and at the other side to the fan

blades whereby the fan induces a flow of cooling fluid through and around the bearing and out through the inter-blade spaces of the fan.

4. An axial flow gas-turbine comprising a rotor, a shaft for the rotor, an anti-friction bearing about the shaft adjacent the rotor, a centrifugal fan having blades lying along and shrouded at that side by the face of the rotor next said

bearing, a shroud on the outer side of said fan blades having a substantially central eye adjacent the bearing whereby the fan induces a flow of air through the bearing and out through the inter-blade spaces of the fan.

LEONARD FREDERICK GEORGE BUTLER.

ERNEST BRIGGS.

EDWARD BOWNESS.