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(54) **SHAFT BEARING FOR A TURBOMACHINE, TURBOMACHINE, AND METHOD OF OPERATING A TURBOMACHINE**

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(57) **ABSTRACT**

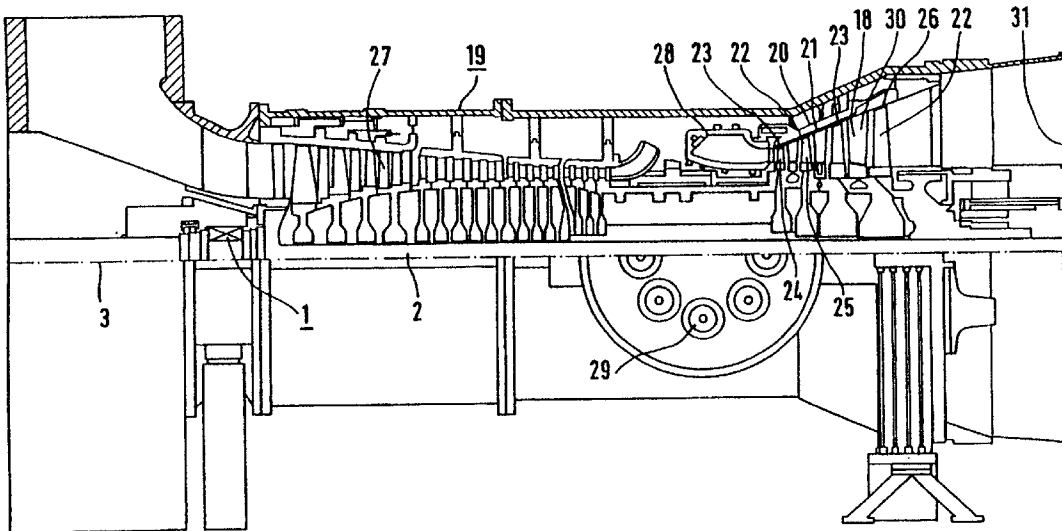
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(63) Continuation of application No. PCT/DE99/03488, filed on Nov. 2, 1999.

The shaft bearing supports the rotor of a turbo-machine which extends along a rotor axis. The shaft bearing has a bearing element with a bearing surface. The bearing surface supports an associated rotor surface and the bearing element can be axially displaced. The invention further relates to a turbo-machine and to a method for operating said turbo-machine.



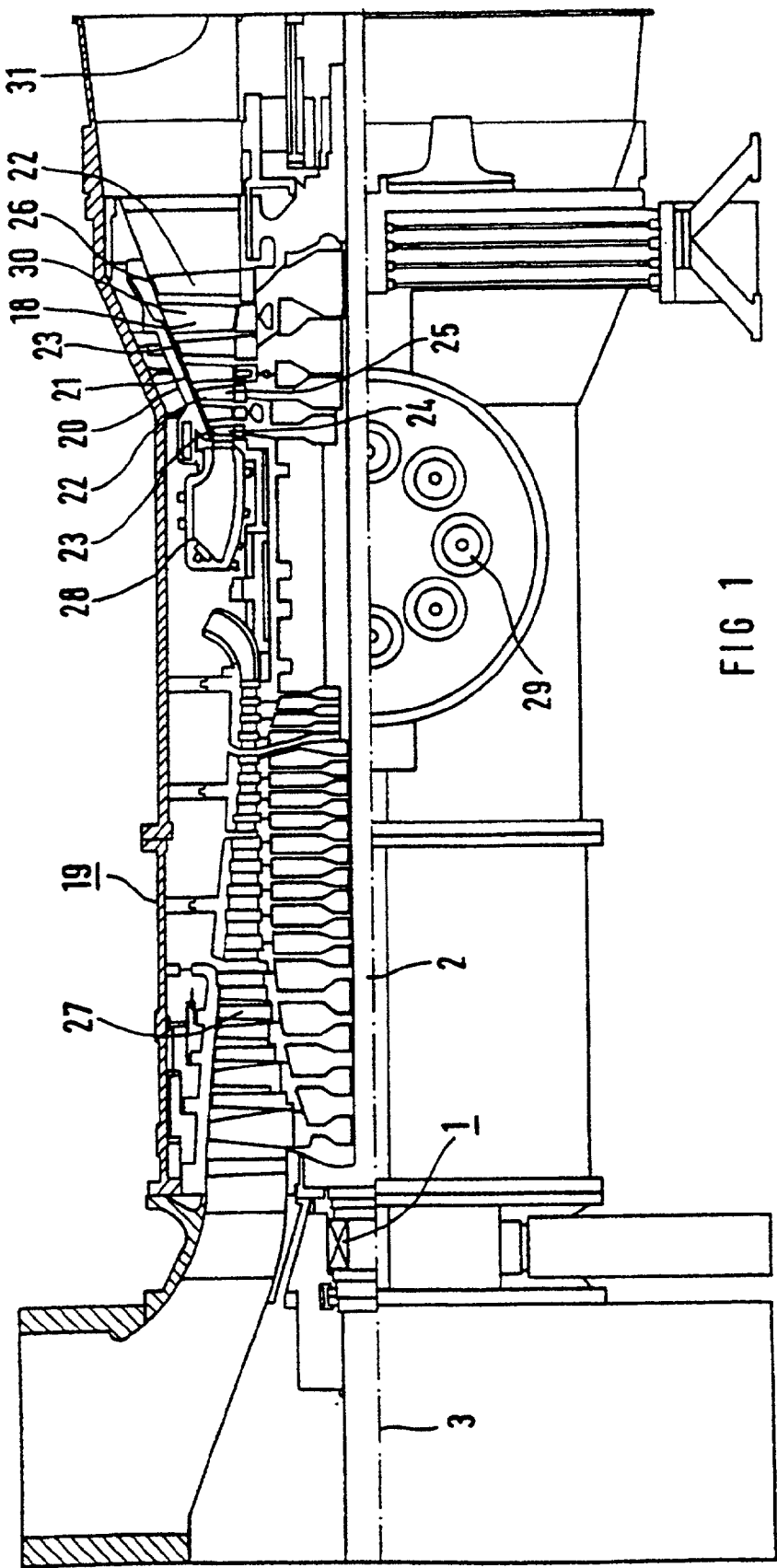
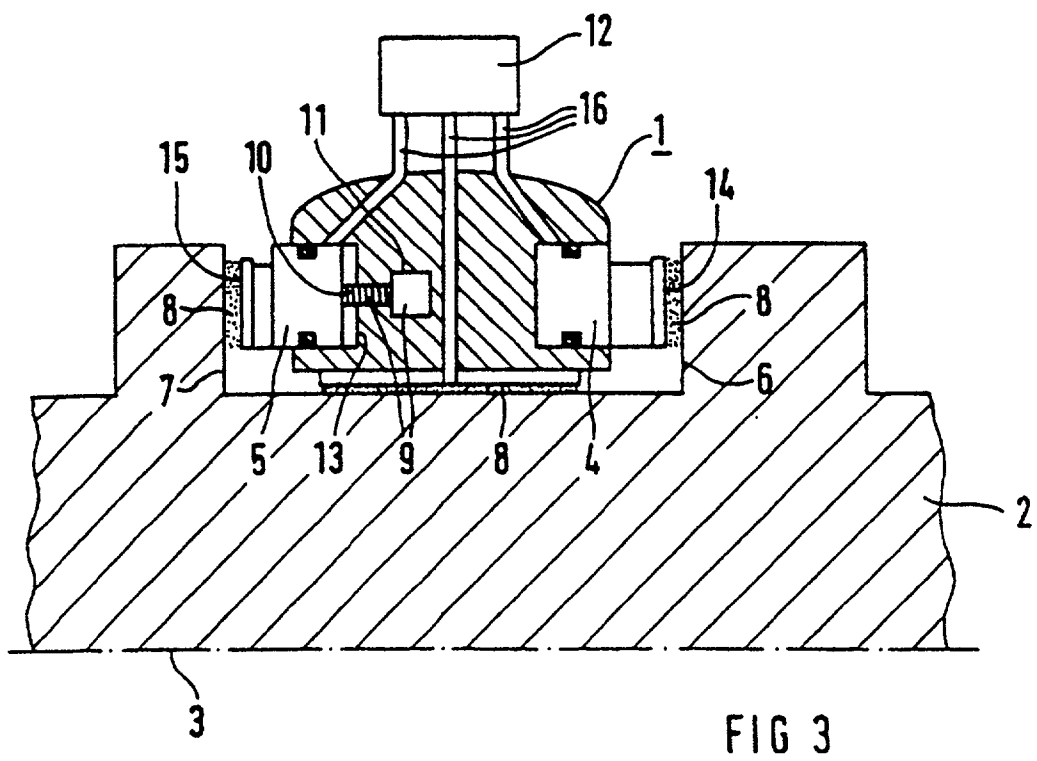
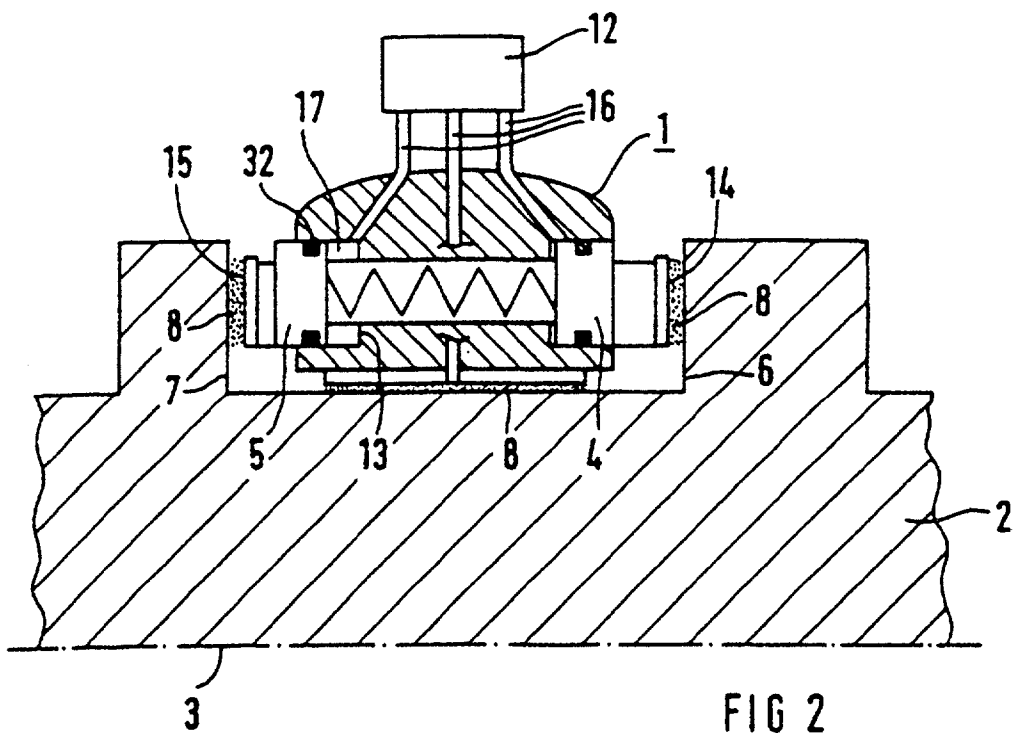
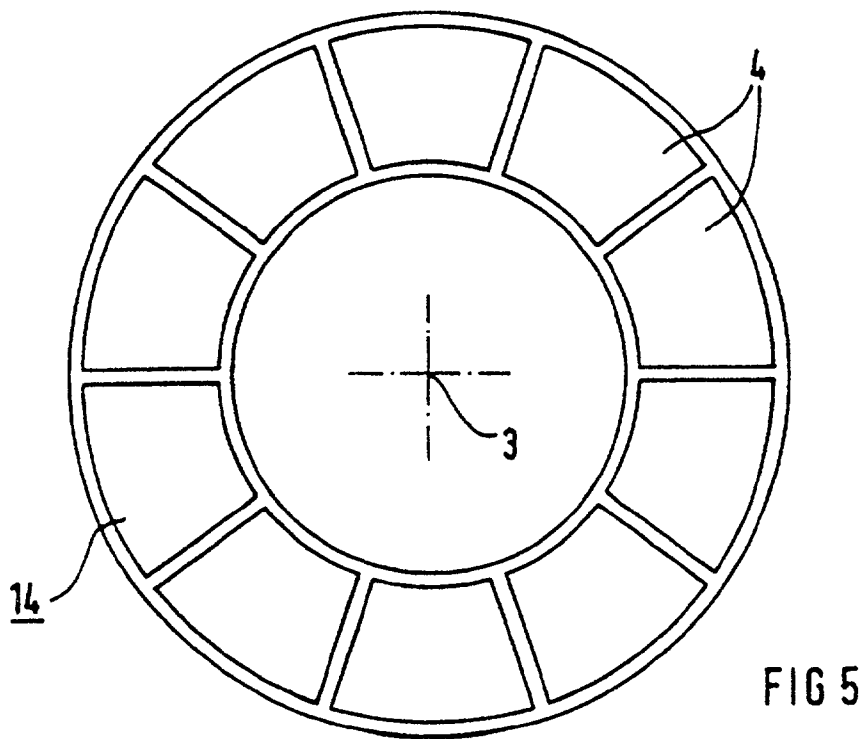
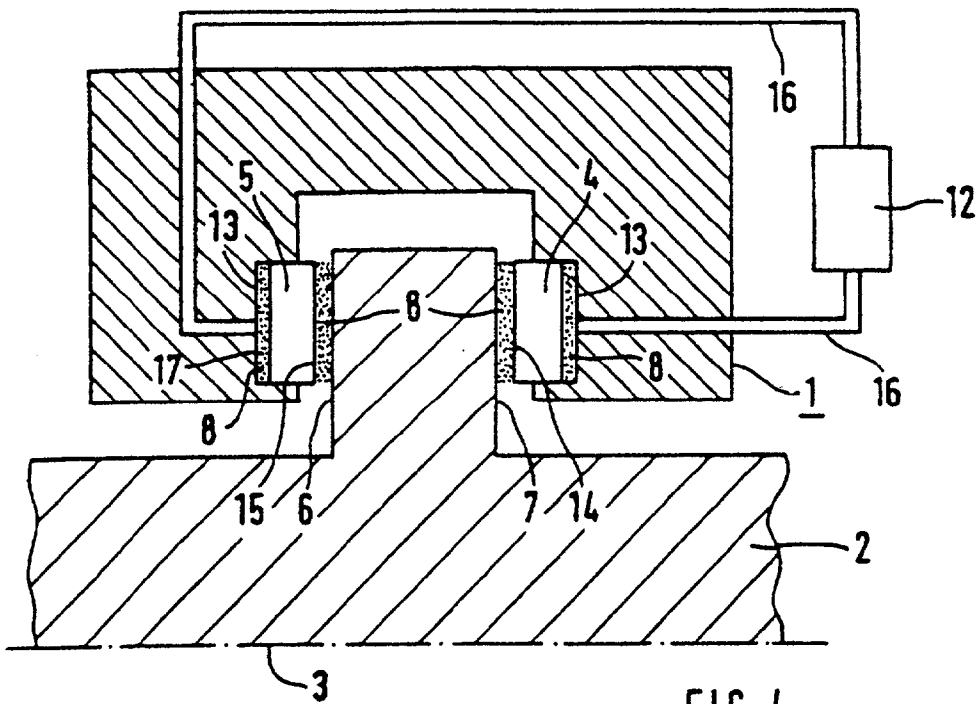


FIG 1





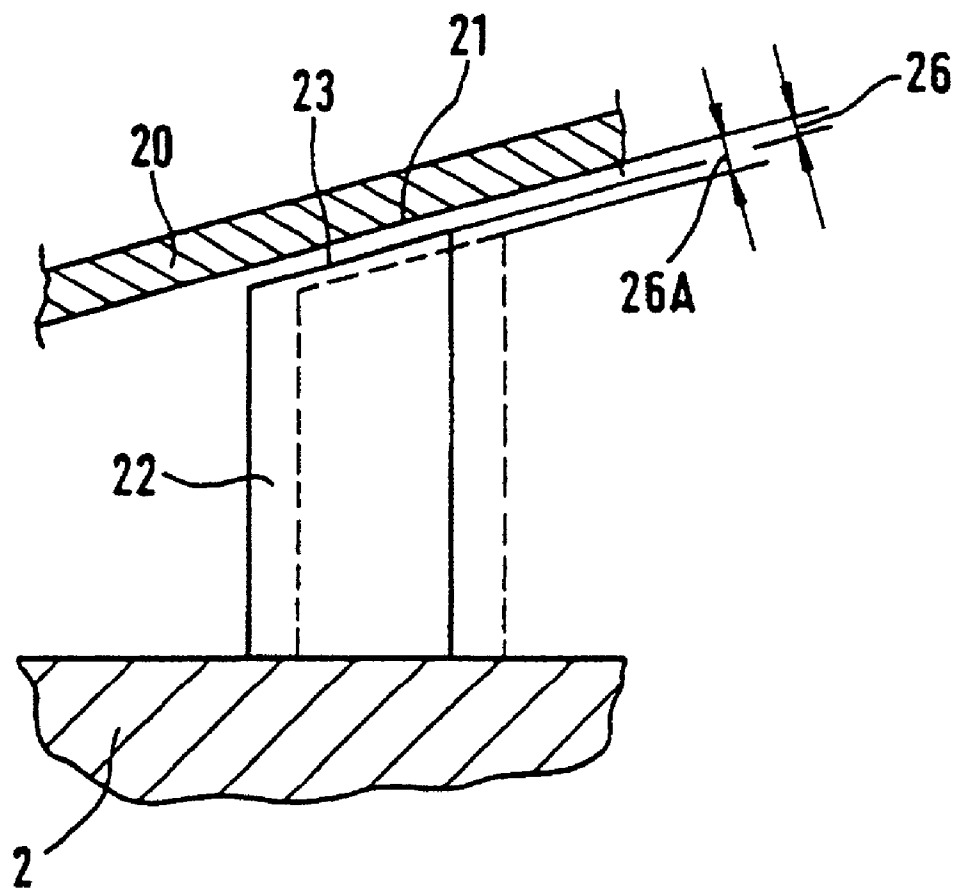


FIG 6

SHAFT BEARING FOR A TURBOMACHINE, TURBOMACHINE, AND METHOD OF OPERATING A TURBOMACHINE

CROSS-REFERENCE TO RELATED APPLICATION

[0001] This application is a continuation of copending International Application No. PCT/DE99/03488, filed Nov. 2, 1999, which designated the United States.

BACKGROUND OF THE INVENTION

FIELD OF THE INVENTION

[0002] The invention lies in the mechanical arts. More specifically, the invention relates to a shaft bearing for the rotor of a turbomachine. The rotor extends along a rotor center line. The shaft bearing has a first bearing element which has a first bearing surface and a second bearing element which has a second bearing surface. In addition, the invention relates to a turbomachine with a rotor which extends along a rotor center line through a casing and in which the casing has a conical inner wall. Rotor blades are arranged on the rotor and these each have a blade tip which faces toward the inner wall and which, in a manner analogous to the inner wall, is conical. The turbomachine has a shaft bearing of the type mentioned above. The invention relates, furthermore, to a method of operating a turbomachine, in which method a displacement of the rotor is carried out relative to the casing.

[0003] Published international PCT application WO 93/20335 describes a method and a configuration for controlling a gap width between the blade tip of a rotor blade and a stationary casing of a rotating machine with a turbine part and a compressor part. There, the control of the gap width takes place in such a way that, during the startup phase of the gas turbine, the shutting down of the gas turbine and load changes of the gas turbine, the gap width is larger than it is during continuous operation of the gas turbine. By this means, the danger of a turbine blade rubbing on the casing during starting, shut-down and load change is reduced. For this purpose, the rotors of the compressor and the turbine are permanently connected together so that they form a single rotor. The casings of the compressor and the turbine are separated from one another and the compressor casing is arranged so that it can be displaced relative to the turbine casing. Due to a displacement of the compressor casing, a displacement of the complete rotor and, therefore, a displacement between the turbine rotor blades and the turbine casing, take place simultaneously.

[0004] U.S. Pat. No. 1,823,310 describes a turbine, in particular a steam turbine, in which means are provided for an axial displacement of the rotor, so that a gap width is provided between the turbine rotor blades and the turbine casing which is greater during the starting and shutting down of the turbine than during the normal operating conditions of the turbine. By this means, a rub or damage to the turbine rotor blades is prevented. The means for displacing the rotor blades act on a thrust bearing. This is connected to the rotor in such a way that when the thrust bearing is displaced axially, the rotor is also displaced. In that configuration, the means comprise a system consisting of gearwheels and racks by means of which the thrust bearing, and therefore the rotor

also, are axially displaced. The same setting principle for the gap width between the rotor blades of a turbine and a turbine casing is employed in French published patent application FR 2 722 836 A1 for a gas turbine with a compressor. There, the gas turbine is supported at a turbine end so that it can be axially moved in a bearing. Further support to the gas turbine takes place at a compressor end by means of a ball bearing, which fixes the complete gas turbine axially. The ball bearing, which is permanently connected to the rotor of the gas turbine, can be axially displaced by means of an appliance so that axial mobility of the rotor relative to the turbine casing is also provided. The ball bearing, and therefore the rotor, can be displaced relative to the turbine casing by a displacement of ± 2 mm.

[0005] U.S. Pat. No. 5,263,817 specifies a centrifugal compressor and a gas turbine with a device for an active gap width control between the rotor blades and the stationary casing. A ball bearing, whose outer race is clamped into the casing in such a way that there is still a small amount of axial mobility, which is limited by a stop, is attached to the rotor. In that configuration, the outer race of the bearing is clamped between the stop and an electromagnetic drive, each of which is permanently connected to the casing. The electromagnetic drive has an electromagnet which is axially adjacent to a radial ferromagnetic disk permanently connected to the rotor. The disk is attracted to a greater or lesser extent, depending on the strength of the magnetic field generated by the electromagnets, so that the axial position of the complete rotor can be varied. By this means, it is possible to actively control the axial position of a rotor blade, and therefore of the gap width between the rotor blade and the conical turbine casing.

[0006] German published patent application DE 42 23 495 A1 describes a gas turbine in which a rotor displacement device is provided for setting a small blade clearance. The rotor displacement device consists of a two-part suspended casing for accommodating a thrust bearing, two annular support plates, which are fastened to the induction casing and to which pressure cells are attached. By means of these, the complete bearing position, and therefore the position of the rotor, can be set in such a way that an increased clearance can be set in the conical blade duct during the start-up and shut-down phases of the gas turbine.

[0007] A common feature of all the prior art devices for displacing a rotor of a turbomachine is substantial structural complication and the interference with the design of a casing or a rotor, as well as the correspondingly increased fault susceptibility.

SUMMARY OF THE INVENTION

[0008] The object of the present invention is to provide a shaft bearing for a turbomachine which overcomes the above-noted deficiencies and disadvantages of the prior art devices and methods of this general kind, and which permits simple axial displacement of the rotor. Further objects of the invention are to provide a turbomachine with an axially displaceable rotor and a method of operating a turbomachine with an axially displaceable rotor.

[0009] With the above and other objects in view there is provided, in accordance with the invention, a turbomachine, comprising:

[0010] a casing with a conical inner wall;

[0011] a rotor extending through the casing along a rotor center line through the casing, the rotor carrying a plurality of rotor blades each having a blade tip facing towards the inner wall and having a conicity substantially corresponding to the inner wall, whereby a radial gap is formed between each the blade tip and the inner wall;

[0012] a shaft bearing having at least one axially displaceable bearing element formed with a bearing surface immediately adjacent a rotor surface and being capable of axially displacing the rotor surface; and

[0013] a mechanical displacement device connected to the bearing element and configured to displace the bearing element and the rotor, and to compensate for an increase in the radial gap upon heating during an operation of the turbomachine.

[0014] In other words, the objects are achieved with the shaft bearing which has a bearing element with a bearing surface, the bearing element with the bearing surface serving as bearing for a rotor surface and being axially displaceable in order to displace the rotor.

[0015] The invention is premised on the realization that a structural redesign of a shaft bearing in such a way that the displacement of a rotor takes place by means of the shaft bearing can be achieved without an essential design change to the rotor or to a casing of a turbomachine. According to the invention, this takes place by a bearing element, which serves as bearing for a rotor surface, being axially displaceable. In this case, axially displaceable signifies displacement in the axial direction relative to a fixed point so that, as a result of the bearing arrangement for the rotor surface, the rotor itself can also be axially displaced. In this arrangement, the shaft bearing is axially fixed. A shaft bearing is then preferably arranged between two radial rotor surfaces of the rotor, a displacement of the rotor by means of the shaft bearing being possible without design changes to the rotor and to the casing. For this purpose, at least one bearing element and therefore one bearing surface is axially displaceable. During an axial displacement of this bearing element, an axial displacement of the rotor takes place simultaneously.

[0016] This is particularly advantageous in the case of turbomachines such as compressors, gas turbines, and steam turbines. In this way, flow losses in radial gaps of a turbine, in particular a stationary gas turbine with a conical inner wall, can be reduced. The radial gaps are then free spaces between the radially outer edge of rotor blades attached to the rotor and the opposite casing parts (inner wall). Because of the pressure difference between the pressure side and the suction side of the rotor blades, working fluid flows through these gaps during the operation of the turbine; in the case of a gas turbine, the working fluid is gas. The mass flow through the gaps does not take part in the conversion to work in the rows of rotor blades at an axial distance from one another and therefore reduces the efficiency of the turbine. In a gas turbine, approximately 30% of the flow losses can be caused by gap leakage. This implies a reduction in the efficiency of the gas turbine by up to 4%. The magnitude of the leakage mass flow, and therefore also the level of the flow losses caused, is determined by the gap width of the radial gaps. In the case of gas turbines operating in a

steady-state manner, what then matters is the gap widths which become set during operation in the continuous powered operation condition, i.e. during the set powered operation condition. In what follows, this gap width is designated as the hot gap. The reasons for the presence of these hot gaps are, for example, the deviations produced by manufacturing tolerances and a safety reserve, for example for unusual operating conditions in the case of earthquakes or the like. Approximately half of the hot gap occurring appears due to time-dependent expansions of the individual turbine components; a steady-state condition of the hot gap appears after the turbine has heated through completely, it being possible for the individual turbine components, such as casing parts or rotor blades, to take up very different temperatures in this operating condition of the turbine, which different temperatures can inter alia also cause distortions of the individual components. By means of the shaft bearing specified, it is inter alia possible to carry out an axial displacement of the rotor in a simple manner after a steady-state operating condition has been reached, in particular a powered operation condition of a turbomachine, so that the magnitude of the hot gap can be set to a prescribed value which is as small as possible while taking account, if necessary, of existing manufacturing deviations and safety reserves.

[0017] The shaft bearing preferably has a further bearing surface which is at an axial distance from the other bearing surface. The two bearing surfaces then serve as bearings for respectively different rotor surfaces at an axial distance from one another. Each of the bearing surfaces can then be formed by an individual, in particular annular bearing element or by a plurality of bearing elements. The two bearing surfaces are then respectively configured to be axially displaceable by means of corresponding bearing elements. The shaft bearing is then a thrust bearing, on which there is a loaded side (bearing surface) which accepts the rotor axial thrust, which occurs mainly in this direction, and an unloaded side (the further bearing surface) which, for example, briefly accepts load under transition conditions (starting) or in the case of faults. During the operation of the bearing, both bearing surfaces are then provided with a corresponding lubricant film (oil film) so that they act as corresponding sliding surfaces. A bearing clearance present due to the lubricant film is then preferably of the order of value of some tenths of a millimeter. The bearing clearance occurring can be maintained by a displacement, in particular a unidirectional displacement, of the two bearing surfaces. By this means, a one-sided, smaller bearing clearance on one of the bearing surfaces is avoided so that no additional losses occur. In addition, this avoids an excessively large bearing clearance which, in the case of an alternating axial thrust direction of the rotor, could lead to undesirable and powerful motions of the rotor with high peak accelerations and inertia forces.

[0018] It is likewise conceivable for one bearing surface to be axially displaceable by an axial displacement of one bearing element or a plurality of bearing elements and for the other bearing surface to be axially fixed. In this case, a displacement of the rotor takes place by means of only one bearing surface, by which means an embodiment of the shaft bearing can be achieved with low design and technical supply complication.

[0019] In accordance with an added feature of the invention, the axially displaceable bearing element is an axially displaceable annular piston. A particularly uniform loading

of the shaft bearing and of the rotor surfaces can be achieved by this means. This is, furthermore, particularly favorable if the axially displaceable bearing element can be displaced by hydraulic means because a uniform pressure distribution over the complete periphery of the annular piston is then ensured by this means. It is, however, likewise possible to provide a plurality of axially displaceable bearing elements (bearing pads), which are, in particular, arranged on a circle concentrically surrounding the rotor.

[0020] In an exemplary embodiment described in the following text, the displaceable bearing element is displaceable by a hydraulic system. In that configuration, the bearing element is acted on by a hydraulic fluid, in particular by an oil, so that a displacement of the rotor is ensured even in the case of full-load operation of the turbomachine. The oil volume necessary for the displacement is then preferably kept constant after a certain axial position has been reached. Because, as is known, oil is an essentially incompressible fluid, the axial position of the bearing surfaces, and therefore of the rotor, never changes to any substantial extent, even in the case of fluctuating forces (actual thrust), provided the oil quantity is kept constant. The use of elastic supply lines (hoses) can be avoided and correspondingly rigid line systems can be employed in order to keep the oil volume correspondingly constant. This avoids the axial position of the rotor alternating between the left-hand stop point and the right-hand stop point in the case of alternating thrust forces and constant oil pressure. If the hydraulic fluid, in particular the oil, is enclosed within a spatial region of constant volume, then, should the thrust force change, the opposing force on the bearing surface will also change so that the force equilibrium is maintained. In the case where no active control of the axial position of the rotor is being carried out, a stop is preferably present in both axial directions so that, by means of a corresponding oil pressure, a pressure is present which acts against the axial thrust of the rotor and clearly overcomes the latter. The displacement then preferably takes place by means of two bearing surfaces which can be displaced in the axial direction, the volume of the hydraulic fluid (oil) pressing on the bearing surfaces being changed in such a way that a desired axial position of the rotor is set and the respective volumes of the hydraulic fluid are then kept constant. The constant volumes achieve the effect that the force caused by the oil pressure and acting on the bearing surface is precisely opposite to and equal to the axial thrust of the rotor. For the supply of the hydraulic fluid, use is preferably made of an already available hydraulic supply arrangement in the case of a shaft bearing which provides a sliding bearing by means of a lubricant (hydraulic fluid). For this purpose, a hydraulic system already employed for lifting the shaft at low rotational speeds can, for example, be used, which system is capable of generating correspondingly high pressures. Such a system could therefore also be additionally selected, if required, in the case of normal operation of a turbomachine in order to effect an axial displacement of the rotor. For this purpose, an additional high-pressure line can be led to the shaft bearing. With this supply arrangement, a hydraulic fluid pressure, in particular an oil pressure, of up to 160 bar is available.

[0021] In accordance with a preferred feature of the invention, there is provided a mechanical displacement device for displacing at least one bearing element. This mechanical displacement device preferably has a displacement element, such as a spindle or the like, and a displacement drive. The

displacement drive is preferably an electric motor. Other possibilities for designing the displacement drive can be mechanical displacement drives which, for example, use the rotation of the rotor during operation of the turbomachine or the flow of the working fluid flowing through the turbomachine.

[0022] The shaft bearing is preferably embodied as a sliding bearing in which a film of a lubricant, in particular hydraulic oil, forms between the bearing surface and the rotor surface. Such a bearing is particularly advantageous for the support of a heavy rotor such, for example, as that employed in stationary gas turbines for the generation of electrical current.

[0023] The bearing surfaces, and therefore the rotor, can preferably be displaced by between 0.5 mm and 5 mm. This displacement is preferably provided in one direction so that the gap width between rotor blades of the turbomachine and the inner wall of the casing of the turbomachine is reduced during normal operation of the turbomachine.

[0024] The shaft bearing preferably has a distance element, for example a stop, by means of which the maintenance of a specified minimum distance between the bearing surfaces is ensured. This is particularly advantageous when a displacement of the rotor against the resultant force acting on the rotor, caused for example by the working fluid flowing through the turbomachine, takes place due to the bearing element. In such a case, the distance element ensures that the rotor takes up an axial position which it would have taken up even without an axial displacement due to the bearing element even in the case of a failure of the hydraulic supply or of the displacement drive. This ensures the operational safety of the turbomachine even in the case of a failure of the axial displacement of the bearing element. Due to the axial thrust occurring in a gas turbine, the rotor is pressed back into its initial position by the gas forces in this case.

[0025] The object directed toward a turbomachine is achieved by one which has a casing with an inner wall which extends conically in the axial direction and in which the rotor guided through the casing has rotor blades whose blade tips facing toward the inner wall extend conically in a manner analogous to the inner wall. The turbomachine then has a shaft bearing which is adjacent to a radial rotor surface, is preferably arranged between two radial rotor surfaces and has at least one axially displaceable bearing element for the displacement of the rotor.

[0026] Due to the conical (tapered) contour, at least in some regions, of the inner wall of the casing of the turbomachine, a change in gap occurs when the rotor is displaced relative to the casing. In the turbomachine specified, the relative position of rotor and casing in the continuous condition (steady-state powered operation condition) of the turbine can be changed in such a way that the hot gap is reduced by the proportion which takes account of the transient thermal expansions. Such transient thermal expansions occur in the turbomachine during the period before all the components of the turbomachine have permanently taken up their steady-state operating temperatures characteristic of the operating condition and have therefore taken up their corresponding thermal expansions (distortions).

[0027] The turbomachine is preferably a gas turbine, an aircraft engine turbine, or a stationary gas turbine for the

generation of electrical energy. A stationary gas turbine can then have an electrical output of more than 60 MW.

[0028] The turbine of the turbomachine preferably has at least two rows of rotor blades (blade rows) which are at an axial distance from one another, the casing and/or the blade tips being designed in such a way that an axial displacement of the rotor provides approximately the same radial gap for each blade row. For this purpose, the obliquity (conicity) is approximately the same in all turbine stages, i.e. blade rows.

[0029] With the above and other objects in view there is also provided, in accordance with the invention, a method of operating a turbomachine, which comprises:

[0030] providing a turbomachine with a casing having a conical inner wall and a rotor in the casing, the rotor carrying a plurality of rotor blades each having a blade tip facing towards the inner wall and having a conicity substantially corresponding to a conicity of the inner wall and forming a radial gap between each blade tip and the inner wall; supporting the rotor in a shaft bearing having at least one axially displaceable bearing element formed with a bearing surface immediately adjacent a rotor surface; and

[0031] selectively displacing the rotor relative to the casing by displacing the axially displaceable bearing element, and thereby adjusting the radial gap between the blade tip and the inner wall to suit a given operating condition of the turbomachine.

[0032] In other words, the displacement of the bearing element effects a displacement of the rotor relative to the casing, so that a specified radial gap is set between the blade tip and the inner wall to suit the operating condition of the turbomachine. This can take place in an essentially passive manner in such a way that an active control of the gap width is avoided and a specified displacement of the rotor is carried out corresponding to the respective operating condition. Such a displacement can, for example, be realized either by the bearing element being acted on by a hydraulic fluid at a specified pressure, so that a specified displacement value is set, or by the bearing element not being acted upon by pressure. The passive setting of the gap width is therefore based on carrying out either no displacement of the rotor or only a specified displacement of the rotor. It is, of course, also possible to carry out a variable displacement of the rotor by means of corresponding appliances.

[0033] A displacement of the rotor is preferably only carried out when a steady-state operating condition of the turbomachine has been reached with a completely steady-state temperature distribution of the individual components of the turbomachine corresponding to the operating condition. The achievement of such an operating condition, in particular of the normal powered operation condition of the turbomachine, can be determined by specifying a previously determined period, by the measurement of temperatures in the casing, by the measurement of temperature differences, by the measurement of a radial gap occurring as a consequence of the thermal expansions, which radial gap is preferably measured, and by a relative displacement between casing and rotor. A relative expansion between rotor and casing is preferably measured at the shaft bearing acting at least as thrust bearing at opposite ends of the rotor.

[0034] Other features which are considered as characteristic for the invention are set forth in the appended claims.

[0035] Although the invention is illustrated and described herein as embodied in a shaft bearing for a turbomachine, turbomachine and method of operating a turbomachine, it is nevertheless not intended to be limited to the details shown, since various modifications and structural changes may be made therein without departing from the spirit of the invention and within the scope and range of equivalents of the claims.

[0036] The construction and method of operation of the invention, however, together with additional objects and advantages thereof will be best understood from the following description of specific embodiments when read in connection with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

[0037] FIG. 1 is a longitudinal section taken through a gas turbine;

[0038] FIG. 2 is a partial sectional view of a shaft bearing with an hydraulically displaceable bearing element;

[0039] FIG. 3 is a partial sectional view of a shaft bearing with an electromechanically displaceable bearing element;

[0040] FIG. 4 is a detail of a longitudinal section of a turbine with a conical casing;

[0041] FIG. 5 is a diagrammatic sectional view of an axial shaft bearing; and

[0042] FIG. 6 is a partial longitudinal section through a turbomachine with a conical housing.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

[0043] Referring now to the figures of the drawing in detail and first, particularly, to FIG. 1 thereof, there is seen a longitudinal section through a turbomachine 19, which, in the present case, is a gas turbine. The machine has a compressor 27 and the actual turbine 18. A combustion chamber 28 with a plurality of burners 29 is arranged between the compressor 27 and the turbine 18. The gas turbine 19 has a rotor 2, which is manufactured from rotor disks on the tie-rod principle. At the compressor end, the gas turbine 19 has a shaft bearing 1 (see FIG. 2 and FIG. 3). The actual turbine 18 has, in a casing 20, an inner wall 21, which expands conically in the axial direction and which is formed from wall segments and guide vanes 30 with guide-vane platforms. The guide-vane platforms and the wall segments can have, respectively, a different inclination relative to the rotor center line 3 of the rotor 2. Rotor blades 22 are connected to the rotor 2 and these rotor blades 22 are arranged in a total of four rotor blade rows 24, 25, which are at an axial distance from one another. Each rotor blade 22 has a blade tip 23, which faces toward the inner wall 21 and has an obliquity (slope relative to the rotor center line 3) corresponding to the inner wall 21. A radial gap 26 (see FIG. 6) is formed between each blade tip 23 and the associated region of the inner wall 21. During normal operation, a hot gas flows through the turbine 18. The hot gas reaches the turbine 18 from the combustion chamber 28 and emerges from the turbine 18 at a turbine outlet 31.

[0044] A shaft bearing 1, which is configured as a stationary sliding bearing, is shown in longitudinal section in FIG. 2. The shaft bearing 1 encloses the rotor 2 in the peripheral

direction and is arranged axially between two radial rotor surfaces 6, 7. The shaft bearing 1 has two bearing elements 4, 5, which are at an axial distance from one another and have respective bearing surfaces 14, 15. The bearing surface 15 of the bearing element 5 is immediately adjacent to the rotor surface 7 and is separated from the latter by a film of a hydraulic fluid (hydraulic oil) 8. Similarly, the bearing surface 14 is separated from the rotor surface 6 by hydraulic fluid 8. In addition, a film of hydraulic fluid 8 is present in the peripheral direction between the rotor 2 and the shaft bearing 1. The shaft bearing 1 is, in this arrangement, a thrust bearing and a journal bearing. The shaft bearing 1 can, of course, be configured as a thrust bearing, in which case a separate journal bearing can be provided. The bearing element 5 can be displaced axially, an oil space 17, into which hydraulic oil can be fed under high pressure, being arranged in the shaft bearing 1 for axial displacement, so that an axial displacement of the bearing element 5 is achieved. The bearing element 5 has, toward the oil space 17, respective sealing rings 32 on an internal periphery (inner diameter) and on an external periphery (outer diameter). The bearing element 5 is preferably configured as an annular piston. The bearing element 4 is preferably likewise configured to be axially displaceable. Respective supply lines 16 for the hydraulic fluid 8, which are connected to a hydraulic supply system 12, lead to the oil space 17, to the bearing element 4, and to the external periphery of the rotor 2. The hydraulic supply system 12 has a non-illustrated reservoir for hydraulic fluid 8 and corresponding, non-illustrated hydraulic pumps for generating a high pressure and for supplying hydraulic fluid to the bearing surfaces 6, 7 and to the external periphery of the rotor 2.

[0045] In this configuration, the hydraulic supply system 12 is preferably configured in such a way that hydraulic fluid can be fed at a corresponding pressure to the bearing elements 4 and 5 so that an axial displacement of the rotor 3 is achieved. After achievement of the axial displacement of the rotor 2, it is possible to keep the volume of the hydraulic fluid acting on the bearing elements 4 and 5 constant in each case by means of the hydraulic supply system 12 or, if appropriate, by means of a different device, for example by means of one or more shut-off valves. This achieves the effect that, due to the incompressibility of the hydraulic fluid, a respectively opposite and equally large counterforce is generated in the shaft bearing 1 even when there are changes to the axial thrust of the rotor 2; the rotor 2 therefore remains in the desired axial position.

[0046] A further embodiment of a shaft bearing 1 is shown in FIG. 3, likewise in longitudinal section. As compared with the embodiment of FIG. 2, this embodiment does not provide hydraulic displacement of the bearing element 5 but, rather, a displacement of the bearing element 5 by electro-mechanical means. With respect to the remaining design of the shaft bearing 1 of FIG. 3, reference is had to the description of FIG. 2. A displacement element 10, in particular a spindle which can be moved in the axial direction by a displacement drive 11, an electric motor in this case, acts on the bearing element 5 within the shaft bearing 1. Together with further non-illustrated components, such as an electrical supply with corresponding electrical lines, the displacement element 10 and the displacement drive 11 form a mechanical displacement device 9 for the axial displacement of the bearing element 5.

[0047] A distance element 13 (see FIG. 2 or 3), here configured as a stop, is provided in the shaft bearing 1. An axial displacement of the bearing element 5 in the direction of the bearing element 4 is limited by the distance element 13. By this means, an axial movement of the rotor 2 in the direction of the bearing element 4 is also limited. This ensures that no displacement (not caused by pure thermal expansions) of the rotor 2 in the direction toward the turbine outlet 31 occurs and leads to a widening of the radial gap and therefore to higher efficiency losses. Even in the case of a failure of the hydraulic supply system or of the displacement device 9, therefore, the radial gap 26 is not larger than that in the case of a gas turbine 19 which does not execute any compensation of the radial gap 26 in consequence of thermal expansions of the rotor 2.

[0048] A further embodiment of a shaft bearing 1, which is configured as a sliding thrust bearing, is shown in FIG. 4. As compared with the embodiments shown in FIGS. 2 and 3, the shaft bearing 1 encloses an annular shaft region which extends in the radial direction and forms the two rotor surfaces 6 and 7. The two bearing surfaces 14 and 15 are respectively adjacent to the two rotor surfaces 6 and 7 and are respectively kept at a distance from the rotor surfaces 6 and 7 by a corresponding lubricant, in particular hydraulic oil. With respect to the further mode of operation and design configuration of the shaft bearing 1, reference should be had to the statements with respect to the embodiments of FIGS. 2 and 3.

[0049] A shaft bearing 1, which has an essentially annular bearing surface 14, is shown in FIG. 5 as diagrammatic cross section. The bearing surface 14 is formed by a plurality of bearing elements 4, bearing pads. In this configuration, the bearing elements 4 can each be displaced individually in the axial direction or can be moved in the axial direction in groups or all together by means of an annular force transmission element, which is not shown for reasons of clarity. It is obviously possible for the bearing surface 14 to be formed by a single annular bearing element.

[0050] A detail of a turbomachine 19 with conically expanding casing 20 (i.e., a taper casing) is shown in FIG. 6 as a longitudinal section. A rotor blade 22 is shown, as an example, on a rotor 2. Its blade tip 23 is embodied with the same obliquity in a manner analogous to the inner wall 21 of the casing. The rotor blade 22 shown by a dotted line corresponds to an operating condition of the turbomachine 19 in which a thermal expansion of the rotor 2 has taken place. Due to the thermal expansion, a relatively large radial gap 26A has appeared between the blade tip 23 and the inner wall 21, through which radial gap 26A, flow losses occur in the turbomachine 19 and cause a reduction in the efficiency. The rotor blade 22 shown by a full line represents an operating condition of the turbomachine 19 in which a displacement of the rotor 2 has been carried out by means of a shaft bearing 1 in order to reduce the radial gap 26, as shown in FIGS. 2 or 3. The radial gap 26 is then distinctly narrower than the radial gap 26A with the non-displaced rotor 2. A reduction in the flow losses in the radial gaps 26 of the turbomachine 19 is achieved by the displacement of the rotor 2 by means of the shaft bearing 1 with an axially displaceable bearing element 5. This method is particularly effective for reducing the flow losses in the case of stationary gas turbines, which are operated in a powered operation condition over a long period.

We claim:

1. A turbomachine, comprising:
 - a casing with a conical inner wall;
 - a rotor extending through said casing along a rotor center line through said casing, said rotor carrying a plurality of rotor blades each having a blade tip facing towards said inner wall and having a conicity substantially corresponding to said inner wall, whereby a radial gap is formed between each said blade tip and said inner wall;
 - a shaft bearing having at least one axially displaceable bearing element formed with a bearing surface immediately adjacent a rotor surface and being capable of axially displacing said rotor surface; and
 - a mechanical displacement device connected to said bearing element and configured to displace said bearing element and said rotor, and to compensate for an increase in said radial gap upon heating during an operation of the turbomachine.
2. The turbomachine according to claim 1, wherein said axially displaceable bearing element is an axially displaceable annular piston.
3. The turbomachine according to claim 1, wherein said mechanical displacement device comprises a displacement element and a displacement drive.
4. The turbomachine according to claim 3, wherein said displacement drive includes an electric motor.
5. The turbomachine according to claim 1, wherein said bearing surfaces are displaceably disposed for displacement by between 0.5 mm and 5 mm.
6. The turbomachine according to claim 1, wherein said bearing is a rotor bearing for stationary operation.
7. The turbomachine according to claim 1, wherein said rotor blades are disposed in at least two rows of rotor blades at an axial distance from one another, and wherein said casing and said blade tips are configured such that an axial displacement of said rotor provides an equal radial gap for each of said rows.

8. A method of operating a turbomachine, which comprises:

providing a turbomachine with a casing having a conical inner wall and a rotor in the casing, the rotor carrying a plurality of rotor blades each having a blade tip facing towards the inner wall and having a conicity substantially corresponding to a conicity of the inner wall and forming a radial gap between each blade tip and the inner wall;

supporting the rotor in a shaft bearing having at least one axially displaceable bearing element formed with a bearing surface immediately adjacent a rotor surface; and

displacing the rotor relative to the casing by displacing the axially displaceable bearing element, and thereby adjusting the radial gap between the blade tip and the inner wall to suit a given operating condition of the turbomachine.

9. The method according to claim 8, which comprises setting the radial gap on reaching a powered operation condition of the turbomachine.

10. The method according to claim 9, which comprises determining that the powered operation condition of the turbomachine has been achieved during a specified time period.

11. The method according to claim 9, which comprises determining that the powered operation condition of the turbomachine has been achieved by determining the axial gap occurring as a consequence of thermal expansions.

12. The method according to claim 9, which comprises determining that the powered operation condition of the turbomachine has been achieved by determining a relative displacement between the casing and the rotor.

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