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(71) **Applicant: REACTION ENGINES LIMITED** [GB/GB]; Building F5 Culham Science Centre, Abingdon Oxfordshire OX14 3DB (GB).

(72) **Inventor: VARVILL, Richard, Anthony**; c/o Reaction Engines Limited Building F5, Culham Science Centre, Abingdon Oxon OX14 3DB (GB).

(74) **Agent: LLOYD, Robin, Jonathan**; Lacon London 84 Theobalds Road, London WC1X 8NL (GB).

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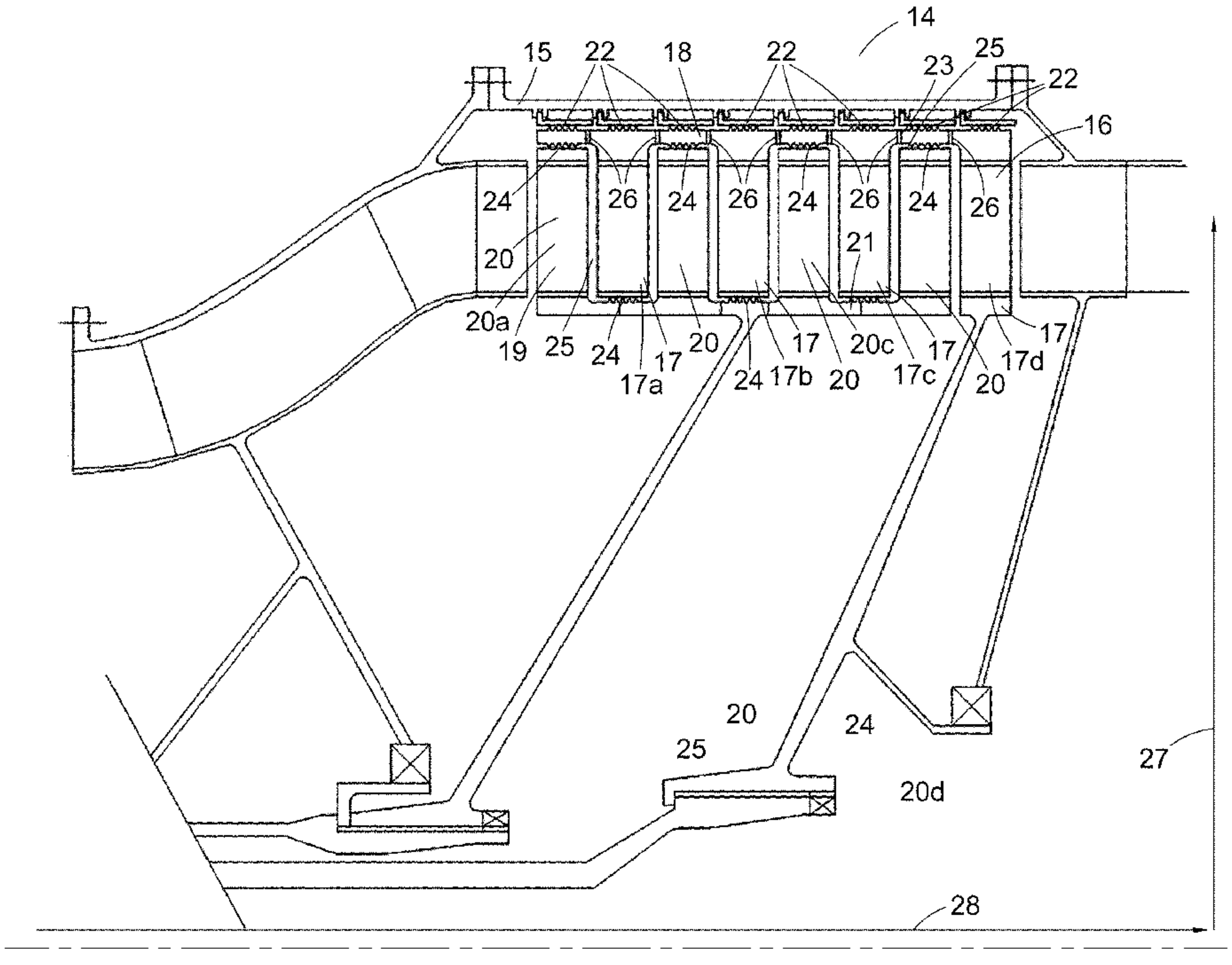

Fig. 3

(57) **Abstract:** A contra-rotating turbine, comprising a housing; a first rotor arranged radially inside the housing; a second rotor arranged radially inside the first rotor; a plurality of first sealing means arranged for at least partially restricting the flow of a working fluid in a downstream direction of the contra-rotating turbine in a first gap between the housing and the first rotor; a plurality of second sealing means arranged for at least partially restricting the flow of a working fluid in the downstream direction of the contra-rotating turbine in a second gap between the first rotor and the second rotor; and one or more fluid flow passages extending through the first rotor, for the flow of said working fluid from the second gap to the first gap.

WO 2021/121994 A1

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CONTRA-ROTATING TURBINE AND METHOD

FIELD

The present disclosure relates to contra-rotating turbines such as of the type which may be used in aerospace applications, automotive applications, industrial applications or other applications. The disclosure also relates to engines and vehicles comprising such contra-rotating turbines, and to a method of reducing tip leakage in contra-rotating turbines.

BACKGROUND

Jivraj, F., Varvill, R., Bond, A. & Paniagua, G., *The Scimitar Precooled Mach 5 Engine. 2nd European Conference for Aerospace Sciences*, describes a counter-rotating turbine, the counter-rotating turbine having blade rows joined to two shafts with opposite rotation directions and different rotation speeds.

Moroz, L., Pagur, P., Govorushchenko, Y. & Grebennik, K., 2009, *Comparison of counter-rotating and traditional axial aircraft low-pressure turbines integral and detailed performances. International symposium on heat transfer in gas turbine systems*, describes a hypersonic air-breathing engine which includes a stator-less contra-rotating turbine.

Varvill, R., Paniagua, G., Kato, H. & Thatcher, M., *Design and testing of the contra-rotating turbine for the Scimitar precooled Mach 5 cruise engine*, describes the design and testing of a contra-rotating turbine.

Such contra-rotating turbines (which may also be referred to in the prior art as being “counter-rotating turbines”), include a low pressure rotor and a high pressure rotor, whereas conventional turbines typically include a single rotor, and a stator. Contra-rotating turbines thus suffer from an additional leak path compared to conventional turbines, namely between the low pressure rotor and the outer casing. This additional leak path is not present in conventional turbines, because there is a stator instead of a (second) low pressure rotor. The additional leak path inherent in contra-rotating turbines is subject to the total pressure difference across the turbine (i.e. to the total pressure difference between the nozzle guide vanes and the last rotor stage of such a turbine), and can lead to high leakage, resulting in a decrease in overall turbine efficiency.

The present disclosure seeks to alleviate, at least to a certain degree, the problems and/or address at least to a certain extent, the difficulties associated with the prior art.

SUMMARY

According to a first aspect of the disclosure, there is provided a contra-rotating turbine, comprising:
a housing;
a first rotor arranged radially inside the housing;
a second rotor arranged radially inside the first rotor;
a plurality of first sealing means arranged for at least partially restricting the flow of a working fluid in a downstream direction of the contra-rotating turbine in a first gap between the housing and the first rotor;
a plurality of second sealing means arranged for at least partially restricting the flow of a working fluid in the downstream direction of the contra-rotating turbine in a second gap between the first rotor and the second rotor; and
one or more fluid flow passages extending through the first rotor, for the flow of said working fluid from the second gap to the first gap.

Advantageously, such a contra-rotating turbine can have reduced leakage. In particular, the pressure drop distribution across the first sealing means can be more balanced, such that there is less leakage in the contra-rotating turbine, resulting in the contra-rotating turbine having an increased overall efficiency (for example by up to 5%, particularly at low rotational speeds). In other words, the pressure drop along the axial length of the contra-rotating turbine can be less uneven, by venting the pressurised working fluid across each of the first sealing means so that the pressure drop of the pressurised working fluid across said first sealing means corresponds to the local stage pressure difference. For example, such a contra-rotating turbine can provide that the otherwise large pressure drop at the most downstream first sealing means is reduced compared with the smaller pressure drops in the other more upstream first sealing means, such that the pressure drops across the first sealing means along the axial length of the contra-rotating turbine are more balanced. Consequently, the loss in turbine efficiency due to the additional leak path present between the housing and the first rotor (i.e. in the first gap, compared with a conventional turbine) is similar to the tip leakage loss. As a result of the aforementioned increase in efficiency, such a contra-rotating turbine can have reduced weight, reduced axial length, and an increased power to weight ratio, and can provide for a reduction or elimination of gyroscopic effects, a reduction in cooling requirements, and a reduced number of required rotor stages. Such a contra-rotating turbine can thus provide for the number of required rotor stages to be reduced for a given value of overall turbine efficiency, or for an increased value of overall turbine efficiency for a given number of rotor stages.

Optionally, the first rotor is configured to rotate in a first direction relative to the housing, and the second rotor is configured to rotate in a second direction relative to the housing, wherein the second direction is opposite to the first direction.

Advantageously, such a contra-rotating turbine can have an increased effective blade speed, compared with a conventional axial turbine having a single rotor and a stator. In particular, as an example, by counter-rotating the first rotor in an opposite direction to the second rotor, the work per moving blade row can be increased by a factor of approximately two and the work per rotor stage can be increased by a factor of approximately four, such that the effective blade speed can be increased, for example, doubled. Consequently, the number of required rotor stages can be reduced, such that the contra-rotating turbine has a reduced mass and an increased efficiency.

Optionally, the first rotor is configured to rotate at a first speed relative to the housing, and the second rotor is configured to rotate at a second speed relative to the housing, wherein the second speed is different to the first speed.

Optionally, the plurality of first sealing means are arranged in the first gap and the plurality of second sealing means are arranged in the second gap.

Optionally, the first rotor is arranged concentrically inside the housing, and the second rotor is arranged concentrically inside the first rotor.

Optionally, the housing, the first rotor and the second rotor share a common longitudinal axis.

Optionally, a liner is arranged in the first gap.

Optionally, the one or more fluid flow passages extend in a substantially radial direction of the turbine.

Advantageously, this can allow for a working fluid to flow from the second gap to the first gap in a substantially radial direction, such that the pressurised working fluid across each of the first sealing means can be vented so that the pressure drop of the pressurised working fluid across said first sealing means corresponds to the local stage pressure difference.

Optionally, the contra-rotating turbine has a longitudinal axis that is perpendicular to the radial direction and extends generally in the downstream direction of the contra-rotating turbine.

Optionally, each of the one or more fluid flow passages comprises a drilled hole through the first rotor.

Optionally, all of the one or more fluid flow passages are equal in size.

Optionally, the contra-rotating turbine comprises a plurality of rotor stages, for example four or five rotor stages.

Optionally, the contra-rotating turbine comprises a plurality of rotor stages, wherein in each of the plurality of rotor stages, at least one of said first sealing means is arranged in the first gap, and at least one of said second sealing means is arranged in the second gap.

Optionally, the contra-rotating turbine comprises a plurality of rotor stages, wherein at least one of the rotor stages comprises at least one of said fluid flow passages.

Advantageously, such a contra-rotating turbine can provide that the pressure of a working fluid is equalised in each of the rotor stages.

Optionally, the contra-rotating turbine comprises a plurality of rotor stages, wherein each of the rotor stages comprises at least one of said fluid flow passages.

Optionally, each of the plurality of rotor stages comprises two of said fluid flow passages, and in each of the plurality of rotor stages, one of said two fluid flow passages is arranged upstream of the at least one first sealing means, and the other one of said two fluid flow passages is arranged downstream of the at least one first sealing means.

Optionally, one or more of the plurality of rotor stages each comprises two of said fluid flow passages, and in each of the plurality of rotor stages, one of said two fluid flow passages is arranged upstream of the at least one first sealing means, and the other one of said two fluid flow passages is arranged downstream of the at least one first sealing means.

Advantageously, this can allow for the pressurised working fluid across each of the first sealing means to be vented so that the pressure drop of the pressurised working fluid across said first sealing means corresponds to the local stage pressure difference. Advantageously, this can provide that the pressure drop distribution across the first sealing means along the axial length of the contra-rotating turbine can be more balanced. In other words, this can provide that the absolute pressure drops across each of the different first sealing means along the axial length of the contra-rotating turbine can be more similar to one another in magnitude. In particular, the fluid flow passages can advantageously connect the main turbine gas passage to the start and end of each of the first sealing means, thus forcing the pressure difference across each of the first sealing means to be equal to the pressure difference across each of the blade rows.

Optionally, along the downstream direction of the contra-rotating turbine, the plurality of first sealing means and the one or more fluid flow passages are arranged alternately with respect to one another.

Optionally, along the downstream direction of the contra-rotating turbine, the plurality of second sealing means and the one or more fluid flow passages are arranged alternately with respect to one another.

Optionally, in each of the plurality of rotor stages, a plurality of said fluid flow passages are arranged spaced apart from one another around a circumferential direction of the contra-rotating turbine.

Optionally, in each of the plurality of rotor stages, the sum of the cross-sectional areas of each of the plurality of fluid flow passages in a direction perpendicular to the radial direction of the contra-rotating turbine is greater than the cross-sectional area of the first gap in a radial plane of the contra-rotating turbine, said radial plane being perpendicular to the downstream direction of the contra-rotating turbine.

In other words, optionally, in each of the plurality of rotor stages, the sum of the cross-sectional areas (in a direction perpendicular to the radial direction of the contra-rotating turbine) of all of the one or more fluid flow passages added together is greater than the cross-sectional area (in a radial plane of the contra-rotating turbine, the radial plane being perpendicular (i.e. normal to) the downstream direction of the contra-rotating turbine) of the first gap.

Advantageously, this can guarantee that the pressure difference across each of the first sealing means corresponds to the local stage pressure difference, to provide a more balanced pressure drop distribution across the first sealing means, such that the contra-rotating turbine has less leakage. The size of each of the one or more fluid flow passages relative to the size of the first gap can be chosen to balance the effects of the mechanical properties of the first rotor being weakened with increasing size of the fluid flow passages, with allowing enough of a working fluid to pass through the fluid flow passages to balance the pressure drop distribution across the first sealing means along the axial length of the contra-rotating turbine. That is, if the one or more fluid flow passages are too large, the mechanical properties of the first rotor might be affected detrimentally. Meanwhile, if the one or more fluid flow passages are too small, then enough working fluid might not be able to flow through the fluid flow passages in order to sufficiently balance the pressure drop distribution across the first sealing means along the axial length of the contra-rotating turbine. These effects may be balanced to determine the optimum sizing of the one or more fluid flow passages relative to the first gap, depending on the circumstances.

Optionally, in each of the plurality of rotor stages, said sum of the cross-sectional areas of each of the plurality of fluid flow passages is at least two times larger than said cross-sectional area of the first gap.

Optionally, in each of the plurality of rotor stages, said sum of the cross-sectional areas of each of the plurality of fluid flow passages is at least five times larger than said cross-sectional area of the first gap.

Optionally, in each of the plurality of rotor stages, the cross-sectional area of each of the plurality of fluid flow passages in a direction perpendicular to the radial direction of the contra-rotating turbine is greater than the cross-sectional area of the first gap in the radial plane of the contra-rotating turbine, said radial plane being perpendicular to the downstream direction of the contra-rotating turbine.

Optionally, the width and/or depth of each of the one or more fluid flow passages in a direction perpendicular to the radial direction of the contra-rotating turbine is greater than the thickness of the first gap in a radial direction of the contra-rotating turbine.

Optionally, said width and/or depth of each of the one or more fluid flow passages is at least approximately two, three, four, five, six or more times larger than said thickness of the first gap.

Optionally, said width and/or depth of each of the one or more fluid flow passages is at least five times larger than said thickness of the first gap.

Optionally, in each of the plurality of rotor stages, each of the plurality of fluid flow passages in the respective rotor stage is sized for allowing the total volumetric flow rate of the working fluid through all of said plurality of fluid flow passages to be greater than the volumetric flow rate of the working fluid in the first gap.

Optionally, the circumferential direction of the contra-rotating turbine is perpendicular to both the radial and downstream directions of the contra-rotating turbine.

Optionally, in each of the plurality of rotor stages, said plurality of fluid flow passages are arranged equally spaced apart from one another around the circumferential direction of the contra-rotating turbine.

Optionally, in each of the plurality of rotor stages, a plurality of said fluid flow passages are arranged in rows, wherein in each of the rows, the fluid flow passages are arranged spaced apart from one another along the downstream direction of the contra-rotating turbine, and wherein the rows are spaced apart from one another around a circumferential direction of the contra-rotating turbine.

Optionally, each of said rows is orientated parallel to the downstream direction of the contra-rotating turbine.

The spacing of the one or more fluid flow passages relative to one another can be chosen to balance the effects of the mechanical properties of the first rotor being weakened with increasing size of the fluid flow passages, with allowing enough of a working fluid to pass through the fluid flow passages to balance the pressure drop distribution across the first sealing means along the axial length of the contra-rotating turbine. That is, if the one or more fluid flow passages are too close together, the mechanical properties of the first rotor might be affected detrimentally. Meanwhile, if the one or more fluid flow passages are too far apart, then enough working fluid might not be able to flow through the fluid flow passages in order to sufficiently balance the pressure drop distribution across the first sealing means along the axial length of the contra-rotating turbine. These effects may be balanced to determine the optimum spacing of the one or more fluid flow passages relative to one another, depending on the circumstances.

Optionally, the cross-sectional area of each of the one or more fluid flow passages perpendicular to the radial direction of the contra-rotating turbine is substantially circular, elliptical, square, rectangular, or slot-shaped.

Optionally, each of the one or more fluid flow passages comprises an aperture, hole, vent, perforation, opening or slot.

Optionally, the working fluid is helium.

Optionally, the first rotor comprises a first shaft and a plurality of first blades arranged on the first shaft, and the second rotor comprises a second shaft and a plurality of second blades arranged on the second shaft.

Optionally, each of the one or more fluid flow passages comprises a drilled hole through the first shaft.

Optionally, the first shaft is generally annular.

Optionally, the housing is generally annular.

Optionally, the first gap is generally annular.

Optionally, the housing comprises an inner radius that is greater than an outer radius of the first shaft, and the first shaft comprises an inner radius that is greater than an outer radius of the plurality of second blades.

Optionally, the plurality of first sealing means are arranged adjacent the first shaft and the housing, and each of the plurality of second sealing means is arranged either adjacent the first shaft and one of the plurality of second blades, or adjacent the second shaft and one of the plurality of first blades.

Optionally, each of the plurality of second sealing means is arranged either between the first shaft and one of the plurality of second blades, or between the second shaft and one of the plurality of first blades.

Optionally, along the downstream direction of the contra-rotating turbine, the plurality of first blades and the plurality of second blades are arranged alternately with respect to one another.

Advantageously, in such a contra-rotating turbine, the previous rotor/blade stage can act as a guiding vane for the next rotor/blade stage.

Optionally, the number of first blades is equal to the number of second blades.

Optionally, the plurality of first sealing means and/or the plurality of second sealing means comprises a plurality of labyrinth seals and/or brush seals and/or any other type of seal.

Optionally, each of the plurality of first sealing means and/or the plurality of second sealing means comprises a labyrinth seal. Each of the labyrinth seals may comprise a plurality of fins extending in a substantially radial direction of the turbine. Each of the labyrinth seals may comprise between two and five fins extending in a substantially radial direction of the turbine. The fins may be configured to at least partially impede/restrict the flow of the working fluid in the downstream direction of the contra-rotating turbine in the first and/or second gaps.

According to a second aspect of the disclosure, there is provided an engine comprising a contra-rotating turbine according to the first aspect of the disclosure.

Advantageously, such an engine can have reduced leakage and improved efficiency in its contra-rotating turbine, as described above in relation to the first aspect of the disclosure.

Optionally, the engine further comprises an air compressor.

Optionally, the air compressor is configured to be driven by the contra-rotating turbine.

Optionally, the working fluid in the contra-rotating turbine is helium.

Advantageously, such an engine can have a reduced weight. This is because due to the differing molecular weights of air and helium, a helium contra-rotating turbine will inherently need to be operated at a faster speed than an air compressor. Hence, otherwise a large number of rotor stages would be required. The engine can thus have an increased power to weight ratio while accounting for the differing properties of air and helium. In other words, such an engine can better compensate for the disparity between the molecular weights of air and helium, or indeed of any other working fluids having differing molecular properties, for example, when the speed of sound in the turbine working fluid is significantly greater than the speed of sound in the compressor working fluid.

Optionally, the engine may comprise any one or more of the optional features of the contra-rotating turbine described above according to the first aspect of the disclosure.

According to a third aspect of the disclosure, there is provided a vehicle, such as an aircraft, flying machine or automobile, comprising a contra-rotating turbine according to the first aspect of the disclosure, or an engine according to the second aspect of the disclosure.

Advantageously, such a vehicle can have reduced leakage and improved efficiency in its contra-rotating turbine, as described above in relation to the first and second aspects of the disclosure.

Optionally, the vehicle may comprise any one or more of the optional features of the contra-rotating turbine described above according to the first aspect of the disclosure, and/or of the engine described above according to the second aspect of the disclosure.

According to a fourth aspect of the disclosure, there is provided a method of reducing tip leakage in a contra-rotating turbine containing a working fluid, the contra-rotating turbine comprising:
a housing;
a first rotor arranged radially inside the housing;
a second rotor arranged radially inside the first rotor;
a plurality of first sealing means arranged for at least partially restricting the flow of a working fluid in a downstream direction of the contra-rotating turbine in a first gap between the housing and the first rotor; and
a plurality of second sealing means arranged for at least partially restricting the flow of a working fluid in the downstream direction of the contra-rotating turbine in a second gap between the first rotor and the second rotor;
the method comprising:
providing one or more fluid flow passages extending through the first rotor, such that at least a portion of the working fluid is permitted to flow from the second gap to the first gap.

Advantageously, such a method can provide for reduced leakage and improved efficiency in a contra-rotating turbine, as described above in relation to the first aspect of the disclosure. In particular, by providing one or more fluid flow passages extending through the first rotor, such that at least a portion of the working fluid is permitted to flow from the second gap to the first gap, the pressurised working fluid across each of the first sealing means can be vented so the pressure drop of the pressurised working fluid across said first sealing means corresponds to the local stage pressure difference, such that the pressure drop along the axial length of the contra-rotating turbine can be more balanced.

Optionally, the working fluid is helium.

Optionally, the step of providing one or more fluid flow passages extending through the first rotor comprises drilling one or more holes through the first rotor.

Optionally, the contra-rotating turbine may comprise any one or more of the optional features of the contra-rotating turbine described above according to the first aspect of the disclosure.

BRIEF DESCRIPTION OF THE DRAWINGS

The present disclosure may be carried out in various ways and embodiments of the disclosure will now be described by way of example with reference to the accompanying drawings in which:

Figure 1 (prior art) shows a schematic cross-sectional view of one stage of a traditional axial turbine;
Figure 2 (prior art) shows a schematic cross-sectional view of one stage of a contra-rotating turbine;
Figure 3 shows an axial cross-sectional view of a contra-rotating turbine;
Figure 4 shows an enlarged view of a portion of Figure 3;
Figure 5 shows a schematic longitudinal cross-sectional view of a shaft of a rotor;
Figure 6 shows a side view of a shaft of a rotor;
Figure 7 shows a side view of a shaft of a rotor; and
Figure 8 shows a plot of pressure against axial distance for a prior art contra-rotating turbine compared with an exemplary contra-rotating turbine according to the present disclosure.

DETAILED DESCRIPTION

Figure 1 (prior art) shows a schematic cross-sectional view of one stage of a traditional axial turbine 1. The traditional axial turbine 1 comprises a stator 3 and a rotor 4 arranged inside an annular housing 2. The rotor 4 comprises a plurality of blades 6 (though only one such blade is illustrated) arranged on a shaft 5. The stator 3 comprises a plurality of vanes (though only one such vane is illustrated) configured to guide and accelerate the flow of a working fluid through the traditional axial turbine 1, while the rotor blades 6 are configured to convert kinetic energy of the moving working fluid into mechanical work on the shaft 5. The flow of the working fluid through the traditional axial turbine 1 is generally parallel to the axis of rotation of the shaft 5.

To illustrate the basic working concept of a contra-rotating turbine, and to highlight the differences between an exemplary contra-rotating turbine and a traditional axial turbine (such as that shown in Figure 1), Figure 2 (prior art) shows a schematic cross-sectional view of one stage of a contra-rotating turbine 7, wherein like reference numerals denote like elements compared with Figure 1. The contra-rotating turbine 7 comprises a first rotor 8 arranged radially inside an annular housing 2, and a second rotor 4 arranged radially inside the first rotor 8. The first rotor 8 comprises a plurality of blades 10 (though only one such blade is illustrated) arranged on an annular shaft 9, while the second rotor 4 comprises a plurality of blades 6 (though only one such blade is illustrated) arranged on a shaft 5. The flow of a working fluid through the contra-rotating turbine 7 is generally radial. The shaft 9 of the first rotor 8 and the shaft 5 of the second rotor 4 are configured to rotate in opposite directions and at different speeds to one another. Along the downstream direction 13 of the contra-rotating turbine 7, the blades 10 of the first rotor 8 and the blades 6 of the second rotor 4 are arranged alternately with respect to one another. In this way, the previous rotor/blade stage can act as a guiding vane (like that of a stator) for the next rotor/blade stage, although there is no traditional stator (such as that in the traditional axial turbine 1 shown in Figure 1) in the contra-rotating turbine 7.

As can best be understood by comparing Figures 1 and 2, the contra-rotating turbine 7 shown in Figure 2 suffers from an additional leak path compared with the traditional axial turbine 1 shown in Figure 1. This additional leak path is located between the housing 2 and the shaft 9 of the first rotor 8, and is not present in the traditional axial turbine 1, because in the traditional axial turbine 1, there is a stator 3 instead of an additional rotor. Specifically, in the contra-rotating turbine 7, there is a first gap 11 between the housing 2 and the shaft 9 of the first rotor 8, and a second gap 12 between the first rotor 8 and the second rotor 4. The traditional axial turbine 1 does not have such a first gap 11.

The additional leak path (in the first gap 11) of the contra-rotating turbine 7 is subject to the total pressure difference across the turbine. Even if this leak path were to be sealed, for example by arranging a number of labyrinth seals (not shown) in the first gap 11, the leakage would be high. This is because the first few (i.e. those closer to the upstream direction) such seals would be subject to very small pressure drops, whereas the last such seal (i.e. the most downstream seal) would be subject to a significantly large pressure drop. Therefore, the pressure drop distribution across the length of the contra-rotating turbine 7 would be inherently unbalanced, and the contra-rotating turbine 7 would therefore suffer from high leakage, and an associated decrease in efficiency.

Figure 3 shows a cross-sectional view of an exemplary contra-rotating turbine 14 according to the present disclosure. An enlarged view is shown in Figure 4. The contra-rotating turbine 14 comprises a first rotor 16 arranged radially inside a generally annular housing 15, and a second rotor 19 arranged radially inside the first rotor 16. The housing 15, the first rotor 16, and the second rotor 19 share a common longitudinal axis 30, such that the first rotor 16 is arranged concentrically inside the housing 15, and the second rotor 19 is arranged concentrically inside the first rotor 16. The common longitudinal axis 30 extends generally in the downstream direction 28 of the contra-rotating turbine 14.

The first rotor 16 comprises a plurality of blades 17 arranged on an annular first shaft 18, while the second rotor 19 comprises a plurality of blades 20 arranged on a second shaft 21. The blades 17 and 20 are arranged alternately with respect to one another along the downstream direction 28 of the contra-rotating turbine 14. That is, a first set 17a of the blades 17 are arranged equally spaced apart from one another around a circumferential direction of the first shaft 18. Second 17b, third 17c and fourth 17d sets of the blades 17 are similarly arranged around the circumferential direction of the first shaft 18, the sets of the blades 17 being spaced apart from one another along the first shaft 18 in the downstream direction 28 of the contra-rotating turbine 14, as shown in Figures 3 and 4. Similarly, the blades 20 are arranged spaced apart from one another around a circumferential direction of the second shaft 21 in sets 20a, 20b, 20c, 20d, the sets of the blades 20 being spaced apart from one another along the second shaft 21 in the downstream direction 28 of the contra-rotating turbine 14. In this way, the pervious rotor/blade stage can act as a guiding vane for the next rotor/blade stage.

In the example shown, there are four rotor stages, such that there are four sets 17a, 17b, 17c, 17d of the blades 17, and four sets 20a, 20b, 20c, 20d of the blades 20. Though, it is also envisaged that there may be any other number of rotor stages, such that there may be any number of sets of the blades 17 and the blades 20, and the number of sets of the blades 17 need not necessarily be equal to the number of sets of the blades 20. For example, there may be five rotor stages, with fives sets of the blades 17 and five sets of the blades 20.

Relative to the housing 15, the first rotor 16 is configured to rotate at a different speed to and in an opposite direction to the second rotor 19.

A plurality of first sealing means 22 are arranged in a first gap 23 between the housing 15 and the first rotor 16, and are configured to at least partially restrict/impede the flow of a working fluid (e.g. helium, though it is also envisaged that the working fluid may also be any other suitable fluid) in the first gap 23 in the downstream direction 28 of the contra-rotating turbine 14, the downstream direction 28 extending parallel to a radial direction 27 of the contra-rotating turbine 14. The first gap 23 is generally annular, and each of the first sealing means 22 extends around the entire circumferential direction 33 of the contra-rotating turbine 14. A liner 29 is arranged in the first gap 23 adjacent the housing 15. Similarly, a plurality of second sealing means 24 are arranged in a second gap 25 between the first rotor 16 and the second rotor 19, and are configured to at least partially restrict/impede the flow of the working fluid in the second gap 25 in the downstream direction 28 of the contra-rotating turbine 14. The second gap 25 has a more complex shape than the first gap 23, in that it extends between the first shaft 18 and the blades 20, between the blades 17 and the blades 20, and between the blades 17 and the second shaft 21. Accordingly, some of the second sealing means 24 are arranged between the first shaft 18 and one of the second blades 20, whereas the remainder of the second sealing means 24 are arranged between the second shaft 21 and one of the first blades 17, as shown in Figures 3 and 4. Each of the second sealing means 24 extends around the entire circumferential direction 33 of the contra-rotating turbine 14.

In the example shown in Figures 3 and 4, the first and second sealing means 22, 24 are labyrinth seals. As shown in Figure 4, in the example, each of the labyrinth seals 22, 24 comprises a plurality of fins 29 extending substantially in the radial direction 27 of the contra-rotating turbine. The fins 29 are configured to at least partially restrict/impede the flow of the working fluid in the first and second gaps 23, 25 by inducing the flow of the working fluid into a vortex-like motion. In this way, the labyrinth seals 22, 24 provide a tortuous path for the working fluid to flow in the downstream direction 28 of the contra-rotating turbine 14 in the first and second gaps 23, 25. In other words, the labyrinth seals 22, 24, particularly the fins 29 thereof, provide that the working fluid has a long and difficult path to escape past/through the seals 22, 24. Thus the seals 22, 24 can restrict/impede any unwanted flow of the working fluid in the downstream direction 28 of the contra-rotating turbine 14 in the first and second gaps 23, 25, to help prevent leakage, particularly in the first gap 23. It is also envisaged that one or more of the first and/or second sealing means may alternatively be any other type of seal, for example brush seals.

Extending through the first rotor 16 are a plurality of fluid flow passages 26. In the example shown in Figures 3 and 4, each of the fluid flow passages 26 comprises a hole drilled through the shaft 18 of the first rotor 16. Though, it is also envisaged that one or more of the fluid flow passages 26 may comprise any other form of aperture, hole, vent, perforation, opening or slot extending through any portion of the first rotor 16, and/or that one or more of the fluid flow passages 26 may be manufactured/formed in any other way.

The fluid flow passages 26 extend substantially parallel to the radial direction 27 of the contra-rotating turbine 14 and provide for the flow of the working fluid from the second gap 25 to the first gap 23; i.e. the fluid flow passages 26 permit at least a portion of the working fluid to flow from between the first rotor 16 and the second rotor 19 to between the first rotor 16 and the housing 15. In other words, each of the fluid flow passages 26 is in fluid communication with the first gap 25 and the second gap 23. In the example shown, the cross-sectional area of each of the fluid flow passages 26 perpendicular to the radial direction 27 of the contra-rotating turbine 14 is circular. Though, it is also envisaged that said cross-sectional area of one or more of the fluid flow passages 26 may be any other shape, such as an ellipses, square, rectangle or slot, or any other suitable shape.

As aforementioned, in the example shown, there are four rotor stages, although it is envisaged that there may be any other number of rotor stages. In each of the rotor stages, there is at least one of the first sealing means 22, at least one of the second sealing means 24, and at least one of the fluid flow passages 26. Along the downstream direction 28 of the contra-rotating turbine 14, in the first gap 23, the first sealing means 22 and the fluid flow passages 26 are arranged alternately with respect to one another, such that the first sealing means 22b, 22c, 22d, 22e, 22f, 22g (see Figure 4) are each arranged between two of the fluid flow passages 26. In other words, the first sealing means 22 and the fluid flow passages 26 are arranged such that all of the first sealing means 22, except for the most downstream 22h and upstream 22a first sealing means, are each arranged between two of the fluid flow passages 26. In this way, each of the first sealing means 22 (except for the most downstream 22h and 22a first sealing means) is in fluid communication with two fluid flow passages 26 (one on either side). This advantageously forces the pressure difference across each of the first sealing means 22 to be equal to the pressure difference across each of the rotor stages/blade rows. It is also envisaged though that the first sealing means 22 and the fluid flow passages 26 may be arranged in any other way with respect to one another in the first gap 23. In each of the rotor stages, each of the fluid flow passages 26 is in fluid communication with its respective first sealing means 22.

In each of the rotor stages, the fluid flow passages 26 are arranged in a set. In other words, the number of sets of fluid flow passages 26 is equal to the number of rotor stages, in the example shown. Though, it is also envisaged that the number of sets of fluid flow passages 26 and the number of rotor stages may not be equal. In the example shown, in each of the sets 26a, 26b, 26c, 26d, 26e, 26f, 26g of fluid flow passages 26, the fluid flow passages 26 are arranged to be equally spaced apart from one another around the circumferential direction 31 of the contra-rotating turbine 14, the circumferential direction being perpendicular to both the radial 27 and downstream 28 directions of the contra-rotating turbine 14. This can best be understood by referring to Figure 5, which shows a schematic axial cross-sectional view of the first shaft 18 of the first rotor 16 (the blades 17 are not shown), showing the arrangement of the fluid flow passages 26 of the first set 26a of fluid flow passages around the circumferential direction 31 of the contra-rotating turbine 14. This is also shown in Figure 6, which shows a side view of the first shaft 18 of the first rotor 16.

It is also envisaged that the fluid flow passages 26 in one or more of the sets 26a, 26b, 26c, 26d, 26e, 26f, 26g may be arranged in any other arrangement or spacing around the circumferential direction 31 of the contra-rotating turbine 14. For example, the fluid flow passages 26 in one or more of the sets 26a, 26b, 26c, 26d, 26d, 26f, 26g may be arranged to be unequally spaced apart from one another around the circumferential direction 31 of the contra-rotating turbine 14.

In terms of the axial spacing of the sets 26a, 26b, 26c, 26d, 26e, 26f, 26g of the fluid flow passages 26 relative to their respective first sealing means 22, it is envisaged that the sets of fluid flow passages 26 may be arranged at any distance relative to/between their respective first sealing means 22.

An alternative exemplary arrangement to that shown in Figure 6 is illustrated in Figure 7. In the arrangement shown in Figure 7, in each of the sets 26a, 26b, 26c, 26d, 26e, 26f, 26g of fluid flow passages 26, the fluid flow passages 26 are arranged in rows 32. In each of the rows 32, the fluid flow passages 26 are arranged spaced apart from one another along the downstream direction 28 of the contra-rotating turbine 14. The rows 32 are each orientated parallel to the downstream direction 28 and are spaced apart from one another around the circumferential direction 31 of the contra-rotating turbine 14. It is envisaged that in each of the sets of fluid flow passages, and/or in each of the rotor stages, the fluid flow passages 26 may be arranged in any other geometrical arrangement/spacing. For example, it is envisaged that in one or more of the sets 26a, 26b, 26c, 26d, 26d, 26f, 26g, there could be any number of fluid flow passages 26 in each of the rows 32, for example two, four, five or more, and that there could be any number of the rows 32.

The geometrical arrangement/spacing of the fluid flow passages 26 relative to one another can be chosen to balance the effects of the mechanical properties of the first rotor 16 being weakened (for example if there are too many fluid flow passages 26, and/or if the fluid flow passages 26 are too close together), with allowing enough of the working fluid to pass through the fluid flow passages 26 in order to sufficiently balance the pressure drop distribution across the first sealing means 22 along the axial length of the contra-rotating turbine 14.

In the example shown, there are four rotor stages, such that there are four sets of the blades 17, four sets of the blades 20, and seven sets of fluid flow passages 26. In the example shown, if *x* is the number of rotor stages, then the number of first sealing means 22 is equal to *2x*, and the number of sets of fluid flow passages 26 is equal to *(2x-1)*. Though, it is also envisaged that there may be any other arithmetic relationship between the number of rotor stages, the number of first sealing means 22, and the number of sets 26a, 26b, 26c, 26d, 26e, 26f, 26g of fluid flow passages 26, depending on the particular circumstances.

Referring now to Figures 4 to 6, the width 33 and depth 34 of each of the fluid flow passages 26 (e.g. the diameter thereof, in the example shown) in a direction perpendicular to the radial direction 27 of the contra-rotating turbine 14 is greater than the thickness 35 of the first gap 23 in the radial direction 27 of the contra-rotating turbine 14. In other words, each of the fluid flow passages 26 is sized to allow a greater volume of the working fluid to flow through them than can flow through/in the first gap 23. Advantageously, this can guarantee that the pressure difference across each of the first sealing means 22 corresponds to the local stage pressure difference, to provide a more balanced pressure drop distribution across the first sealing means 22, such that the contra-rotating turbine 14 has less leakage. The size of each of the one or more fluid flow passages 26 relative to the size of the first gap 23 can be chosen to balance the effects of the mechanical properties of the first rotor being weakened with increasing size of the fluid flow passages 26, with allowing enough of a working fluid to pass through the fluid flow passages 26 to balance the pressure drop distribution across the first sealing means 22 along the axial length of the contra-rotating turbine 14. Preferably, in order to optimally balance these effects, the width 33 and/or depth 34 (or diameter, in the case of a fluid flow passage having a circular cross-section perpendicular to the radial direction 27) of each of the one or more fluid flow passages 26 is at least five times larger than the thickness 35 of the first gap 23. Though, it is also envisaged that other size ratios between the width 33 and/or depth 34 and the thickness 35 may be used. For example, the width 33 and/or depth 23 of each of the one or more fluid flow passages 26 may be at least approximately two, three, four, six or more times larger than the thickness 35 of the first gap 23.

In the example shown, the fluid flow passages 26 are equal in size, i.e. their respective cross-sectional areas perpendicular to the radial direction 27 of the contra-rotating turbine 14 are equal in surface area. Though, it is also envisaged that one or more of the fluid flow passages 26 may be of different sizes to one another.

Advantageously, the contra-rotating turbine 14 can have reduced leakage. In particular, the pressure drop distribution across the first sealing means 22 can be more balanced, such that there is less leakage in the contra-rotating turbine 14, resulting in the contra-rotating turbine 14 having an increased overall efficiency (for example by up to 5%, particularly at low rotational speeds). In other words, the pressure drop along the axial length of the contra-rotating turbine 14 can be less uneven, by venting the pressurised working fluid across each of the first sealing means 22 (by means of the fluid flow passages 26) so that the pressure drop of the pressurised fluid across said first sealing means 22 corresponds to the local stage pressure difference. For example, such a contra-rotating turbine 14 can provide that the otherwise large pressure drop at the most downstream first sealing means 22h is reduced compared with the smaller pressure drops in the other more upstream first sealing means 22a, 22b, 22c, 22d, 22e, 22f, 22g, such that the pressure drops across the first sealing means 22 along the axial length of the contra-rotating turbine 14 are more balanced compared with one another. Consequently, the loss in turbine efficiency due to the additional leak path present between the housing 15 and the first rotor 16 (i.e. in the first gap 23, compared with a conventional turbine) is similar to the tip leakage loss. As a result of the aforementioned increase in efficiency, the contra-rotating turbine 14 can have reduced weight, reduced axial length, and an increased power to weight ratio, and can provide for a reduction or elimination of gyroscopic effects, a reduction in cooling requirements, and a reduced number of required rotor stages. The contra-rotating turbine 14 can thus provide for the number of required rotor stages to be reduced for a given value of overall turbine efficiency, or for an increased value of overall turbine efficiency for a given number of rotor stages. These technical advantages are demonstrated in Figure 8, which shows a plot of pressure against axial distance for a prior art contra-rotating turbine compared with an exemplary contra-rotating turbine according to the present disclosure.

The lower curve of Figure 8 shows a plot of pressure against axial distance (i.e. the position along the downstream direction 28) for the axial turbine 14. The upper curve of Figure 8 shows a plot of pressure against axial distance for an exemplary prior art contra-rotating turbine 7 (such as that shown in Figure 2 and described above). Figure 8 shows that for a contra-rotating turbine with five rotor stages having labyrinth seals as the first sealing means 22 (similar to that shown in Figures 3 and 4, which alternatively has four rotor stages), the presence of the fluid flow passages 26 in the first rotor 16 after each of the first sealing means 22 provides that the pressure drop across each of the rotor stages is much more balanced/even along the length of the contra-rotating turbine. In contrast, in the prior art contra-rotating turbine 7, the pressure drop across each of the first few rotor stages/labyrinth seals is relatively small, whereas the pressure drop across the last/most downstream rotor stage/labyrinth seal is significantly larger. This results in an uneven pressure drop distribution along the length of the prior art contra-rotating turbine 7, resulting in high leakage and an associated decrease in efficiency. On the contrary, by having a much more even/balanced pressure drop distribution along its length, the contra-rotating turbine shown in the upper curve of Figure 8 (which is similar to that shown in Figures 3 and 4, but has five rotor stages rather than four), has a reduced amount of leakage and hence an increased overall turbine efficiency. The advantageous effects of the fluid flow passages 26 on the pressure drop distribution along the length of the contra-rotating turbine 14 may be more beneficial when the number of rotor stages and/or the overall turbine pressure drop is greater.

It should be understood that the disclosure also includes an engine comprising a contra-rotating turbine 14 as described herein. As an example, such an engine may include an air compressor configured to be driven by the contra-rotating turbine 14, wherein the working fluid in the contra-rotating turbine 14 is helium. Advantageously, such an engine can have a reduced weight. This is because due to the differing molecular weights of air and helium, the helium contra-rotating turbine 14 will inherently need to be operated at a faster speed than an air compressor. Hence, otherwise a large number of rotor stages would be required. The engine can thus have an increased power to weight ratio while accounting for the differing properties of air and helium. In other words, such an engine can better compensate for the disparity between the molecular weights of air and helium, or indeed of any other working fluids having differing molecular properties, for example, when the speed of sound in the turbine working fluid is significantly greater than the speed of sound in the compressor working fluid.

Furthermore, it should be understood that the disclosure also includes a vehicle, such as an aircraft, flying machine or automobile, comprising a contra-rotating turbine or an engine as described herein; and a method of reducing tip leakage in a contra-rotating turbine containing a working fluid. The contra-rotating turbine 14 for the method comprises a housing 15; a first rotor 16 arranged radially inside the housing 15; a second rotor 19 arranged radially inside the first rotor 16; a plurality of first sealing means 22 arranged for at least partially restricting the flow of a working fluid in a downstream direction 28 of the contra-rotating turbine 14 in a first gap 23 between the housing 15 and the first rotor 16; and a plurality of second sealing means 24 arranged for at least partially restricting the flow of the working fluid in the downstream direction 28 in a second gap 25 between the first rotor 16 and the second rotor 19. The method comprises providing one or more fluid flow passages 26 extending through the first rotor 16, such that at least a portion of the working fluid is permitted to flow from the second gap 25 to the first gap 23. The step of providing one or more fluid flow passages 26 extending through the first rotor 16 comprises drilling one or more holes through the first rotor 16. Though, it is also envisaged that one or more of the one or more fluid flow passages 26 may be manufactured/formed in any other way. Advantageously, such a method can provide for reduced leakage and improved efficiency in a contra-rotating turbine 14. In particular, by providing one or more fluid flow passages 26 extending through the first rotor 16, such that at least a portion of the working fluid is permitted to flow from the second gap 25 to the first gap 23, the pressurised working fluid across each of the first sealing means 22 can be vented so the pressure drop of the pressurised working fluid across said first sealing means 22 corresponds to the local stage pressure difference, such that the pressure drop along the axial length of the contra-rotating turbine 14 can be more balanced.

Various modifications may be made to the described embodiment(s) without departing from the scope of the invention as defined by the accompanying claims.

CLAIMS

1. A contra-rotating turbine, comprising:

a housing;

a first rotor arranged radially inside the housing;

a second rotor arranged radially inside the first rotor;

a plurality of first sealing means arranged for at least partially restricting the flow of a working fluid in a downstream direction of the contra-rotating turbine in a first gap between the housing and the first rotor;

a plurality of second sealing means arranged for at least partially restricting the flow of a working fluid in the downstream direction of the contra-rotating turbine in a second gap between the first rotor and the second rotor; and

one or more fluid flow passages extending through the first rotor, for the flow of said working fluid from the second gap to the first gap.

2. A contra-rotating turbine as claimed in claim 1, wherein the one or more fluid flow passages extend in a substantially radial direction of the turbine.

3. A contra-rotating turbine as claimed in any of the preceding claims, wherein the contra-rotating turbine comprises a plurality of rotor stages, wherein in each of the plurality of rotor stages, at least one of said first sealing means is arranged in the first gap, and at least one of said second sealing means is arranged in the second gap.

4. A contra-rotating turbine as claimed in any of the preceding claims, wherein the contra-rotating turbine comprises a plurality of rotor stages, wherein at least one of the rotor stages comprises at least one of said fluid flow passages.

5. A contra-rotating turbine as claimed in claim 3, wherein each of the plurality of rotor stages comprises two of said fluid flow passages, and in each of the plurality of rotor stages, one of said two fluid flow passages is arranged upstream of the at least one first sealing means, and the other one of said two fluid flow passages is arranged downstream of the at least one first sealing means.

6. A contra-rotating turbine as claimed in any of claims 3 to 5, wherein in each of the plurality of rotor stages, a plurality of said fluid flow passages are arranged spaced apart from one another around a circumferential direction of the contra-rotating turbine.

7. A contra-rotating turbine as claimed in claim 6, wherein in each of the plurality of rotor stages, the sum of the cross-sectional areas of each of the plurality of fluid flow passages in a direction perpendicular to the radial direction of the contra-rotating turbine is greater than the cross-sectional area of the first gap in a radial plane of the contra-rotating turbine, said radial plane being perpendicular to the downstream direction of the contra-rotating turbine.

8. A contra-rotating turbine as claimed in claim 7, wherein in each of the plurality of rotor stages, said sum of the cross-sectional areas of each of the plurality of fluid flow passages is at least five times larger than said cross-sectional area of the first gap.

9. A contra-rotating turbine as claimed in any of the preceding claims, wherein the cross-sectional area of each of the one or more fluid flow passages perpendicular to the radial direction of the contra-rotating turbine is substantially circular, elliptical, square, rectangular, or slot-shaped.

10. A contra-rotating turbine as claimed in any of the preceding claims, wherein the working fluid is helium.

11. A contra-rotating turbine as claimed in any of the preceding claims, wherein the first rotor comprises a first shaft and a plurality of first blades arranged on the first shaft, and the second rotor comprises a second shaft and a plurality of second blades arranged on the second shaft.

12. A contra-rotating turbine as claimed in claim 11, wherein the plurality of first sealing means are arranged adjacent the first shaft and the housing, and each of the plurality of second sealing means is arranged either adjacent the first shaft and one of the plurality of second blades, or adjacent the second shaft and one of the plurality of first blades.

13. A contra-rotating turbine as claimed in any of the preceding claims, wherein the plurality of first sealing means and/or the plurality of second sealing means comprises a plurality of labyrinth seals and/or brush seals.

14. An engine comprising a contra-rotating turbine as claimed in any of the preceding claims.

15. An engine as claimed in claim 14, wherein the engine further comprises an air compressor.

16. A vehicle, such as an aircraft, flying machine or automobile, comprising a contra-rotating turbine as claimed in any of claims 1 to 13, or an engine as claimed in claim 14 or claim 15.

17. A method of reducing tip leakage in a contra-rotating turbine containing a working fluid, the contra-rotating turbine comprising:

a housing;

a first rotor arranged radially inside the housing;

a second rotor arranged radially inside the first rotor;

a plurality of first sealing means arranged for at least partially restricting the flow of a working fluid in a downstream direction of the contra-rotating turbine in a first gap between the housing and the first rotor; and

a plurality of second sealing means arranged for at least partially restricting the flow of a working fluid in the downstream direction of the contra-rotating turbine in a second gap between the first rotor and the second rotor;

the method comprising:

providing one or more fluid flow passages extending through the first rotor, such that at least a portion of the working fluid is permitted to flow from the second gap to the first gap.

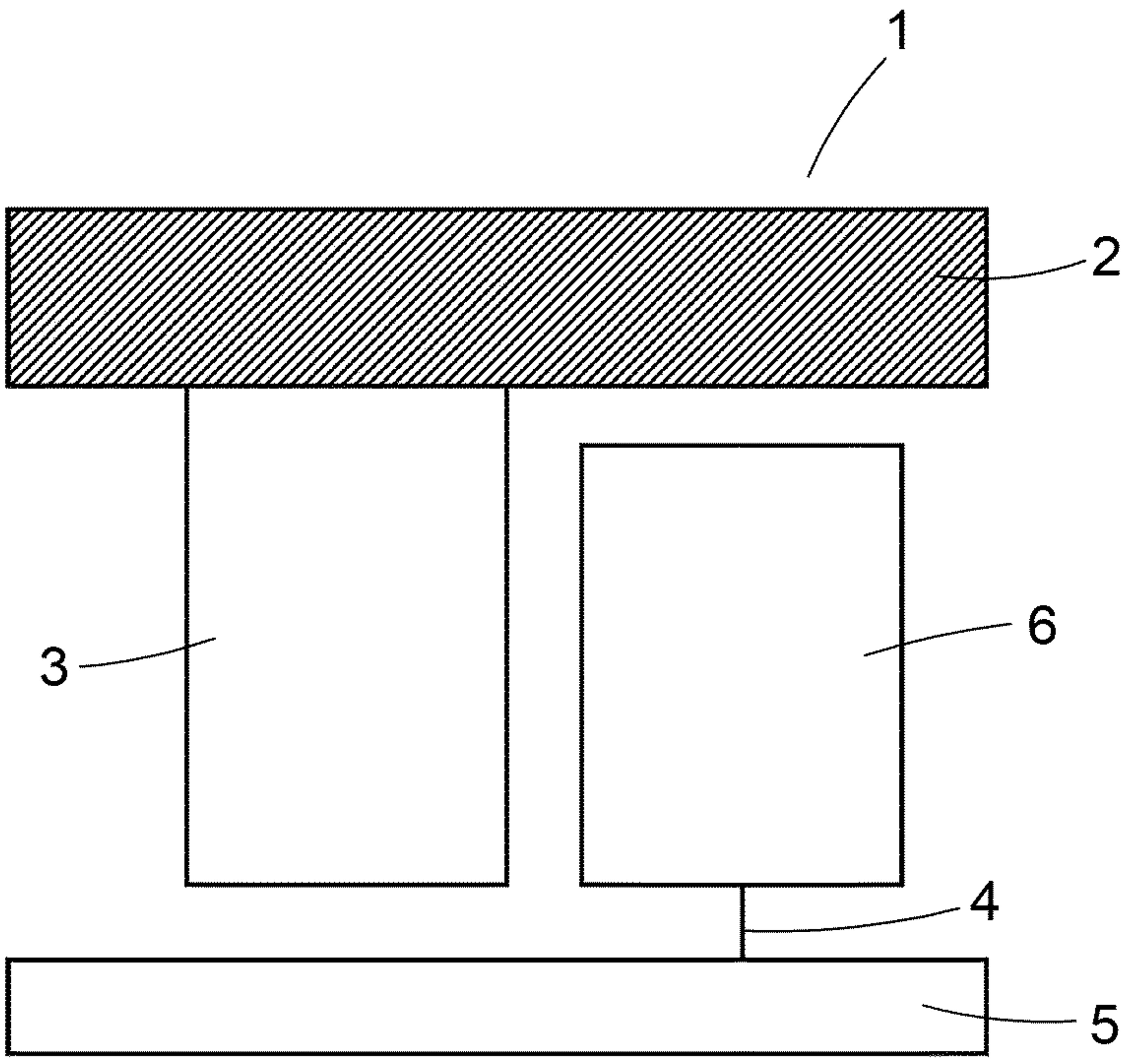

Fig. 1

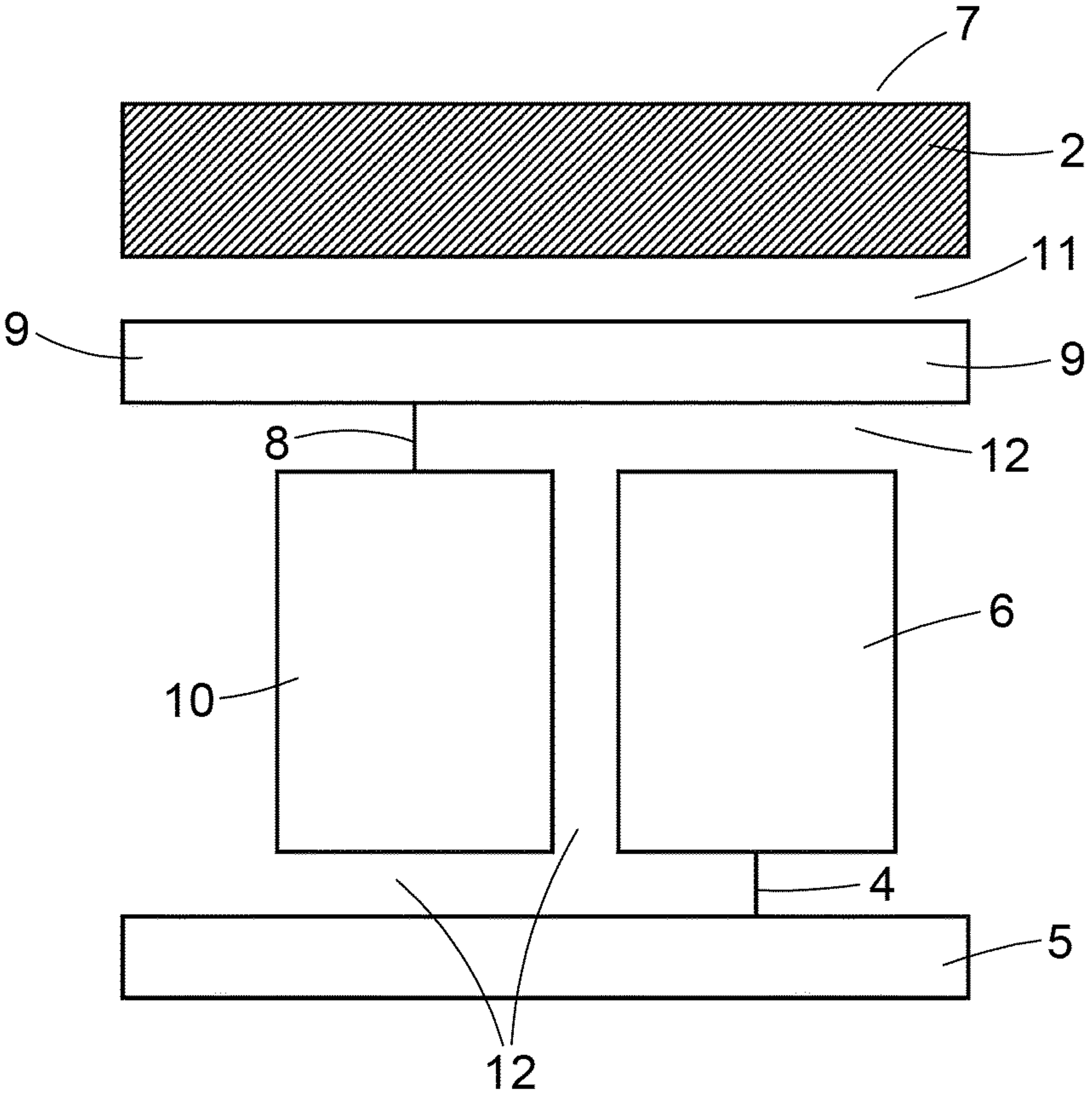

Fig. 2

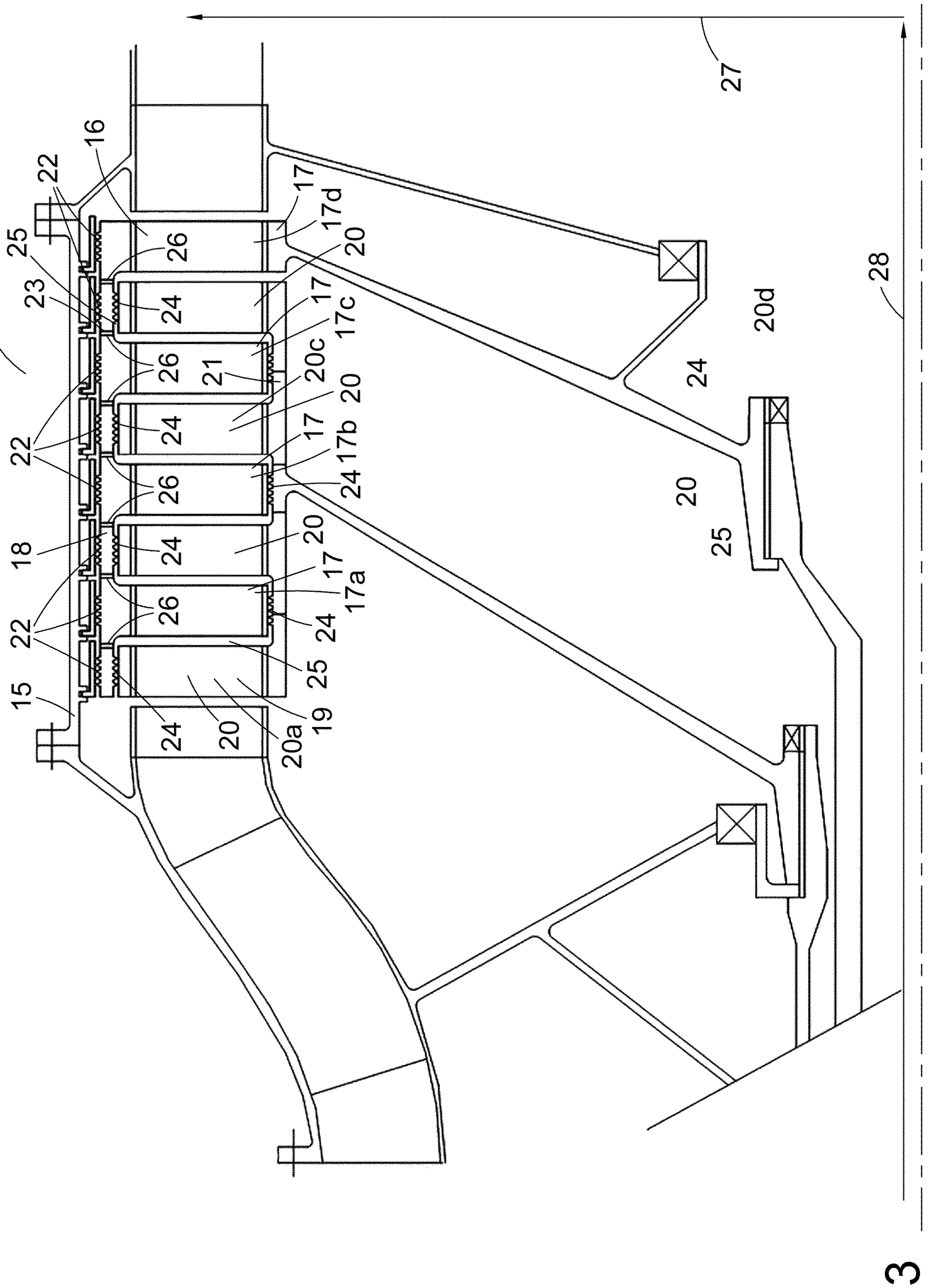
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Fig. 3

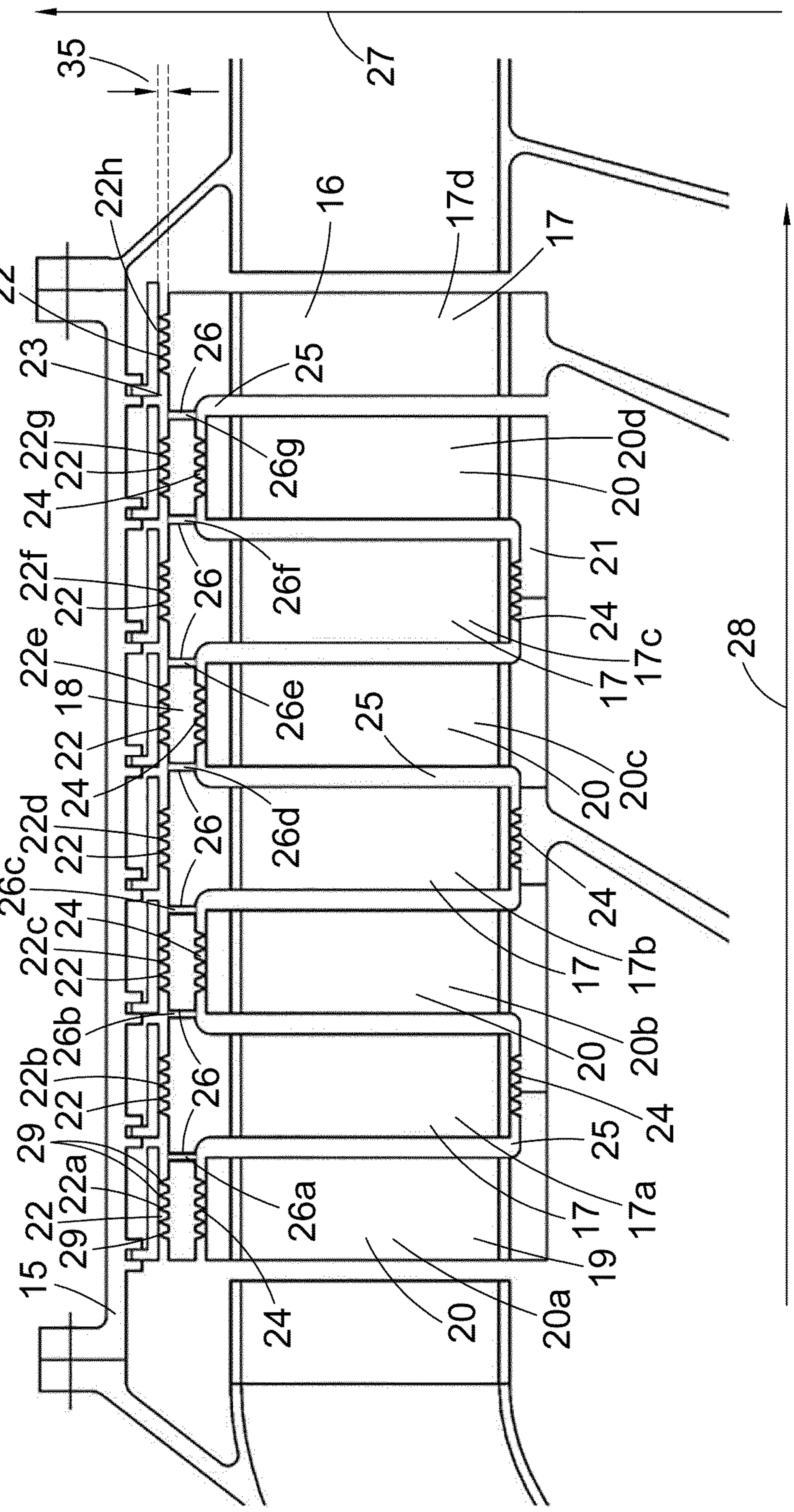
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Fig. 4

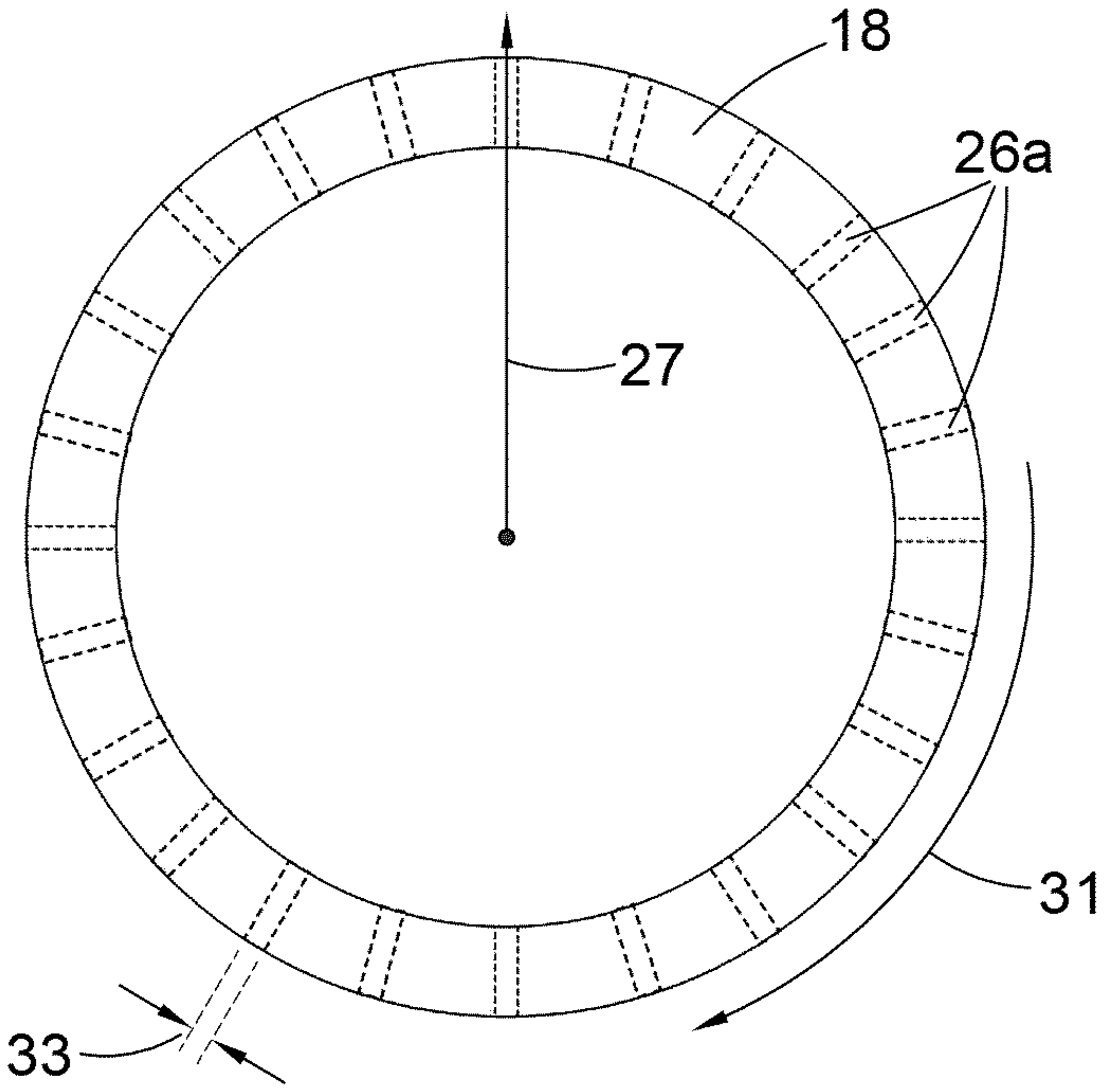
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Fig. 5

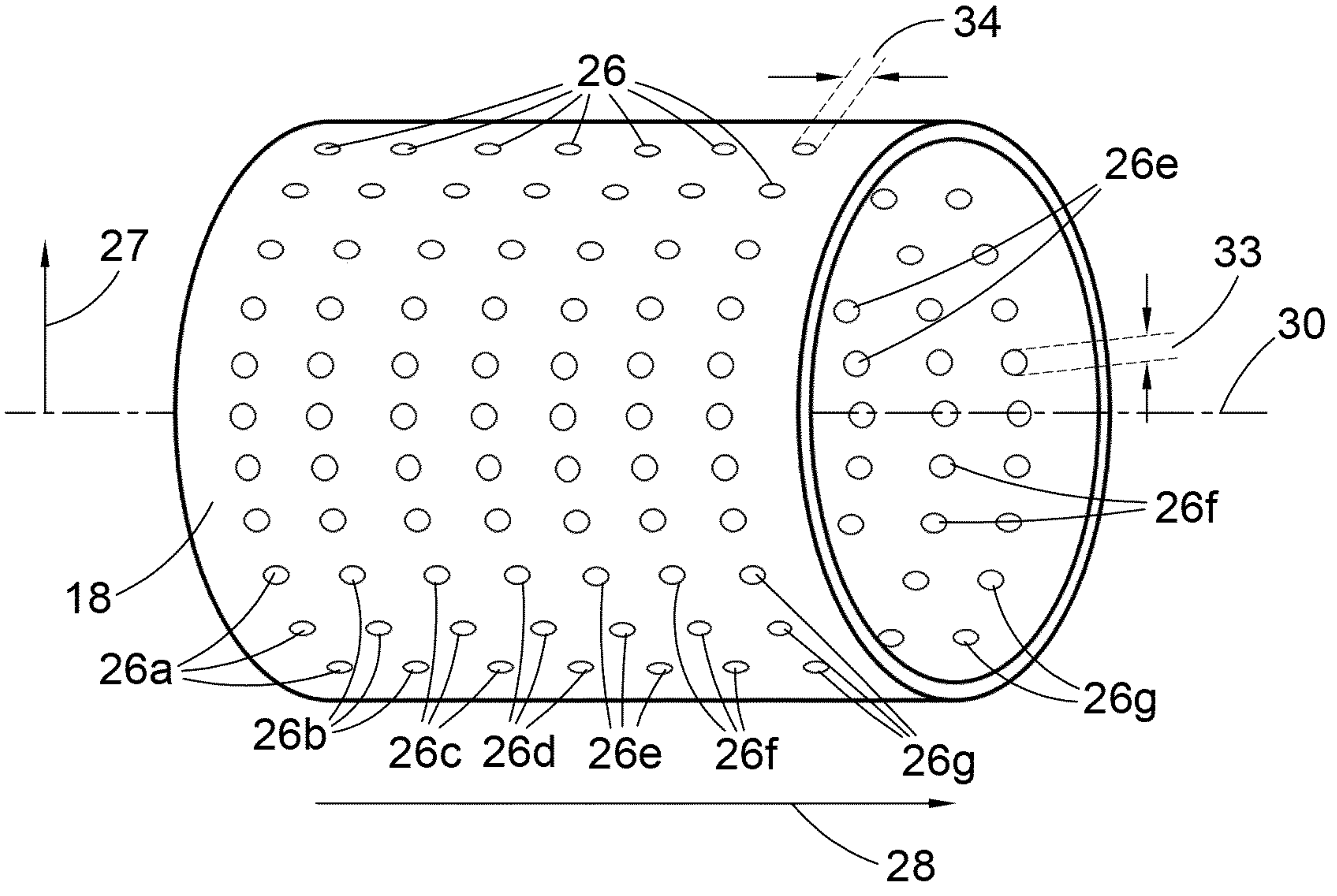
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Fig. 6

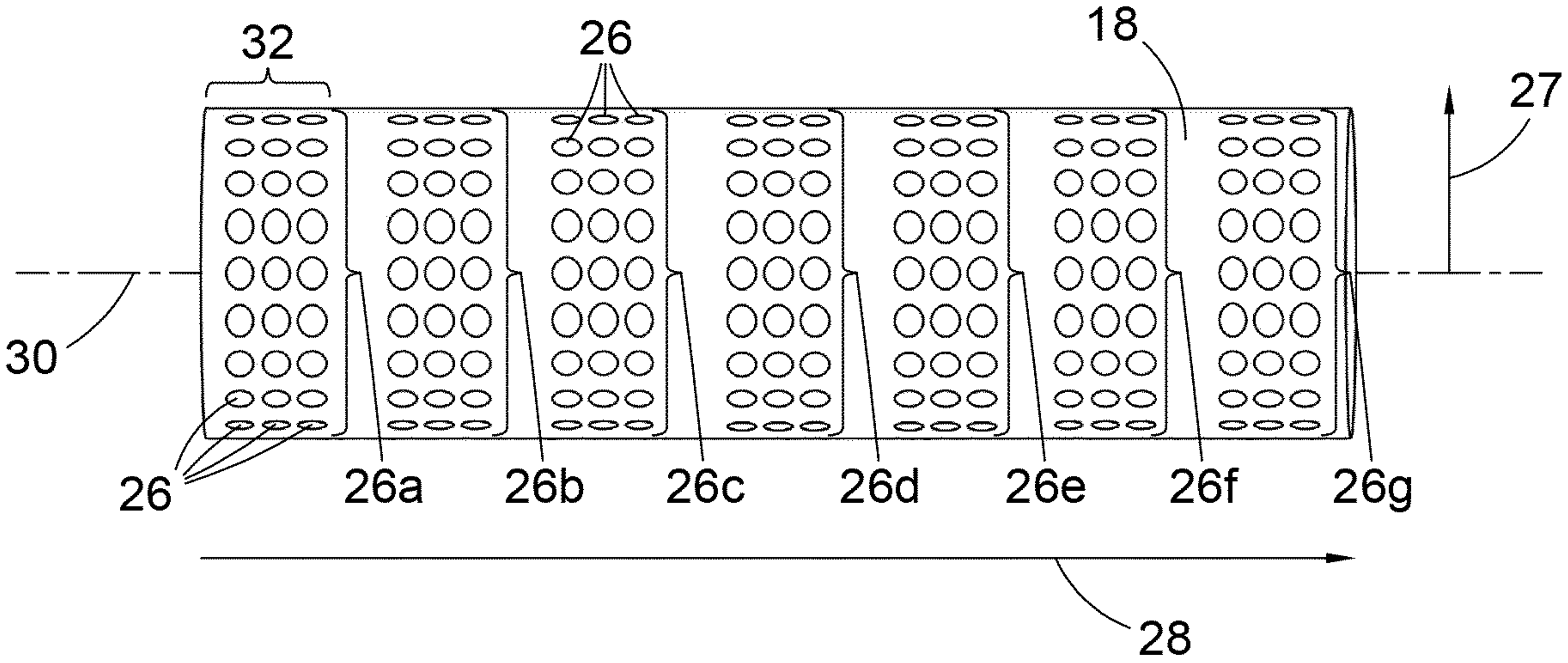
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Fig. 7

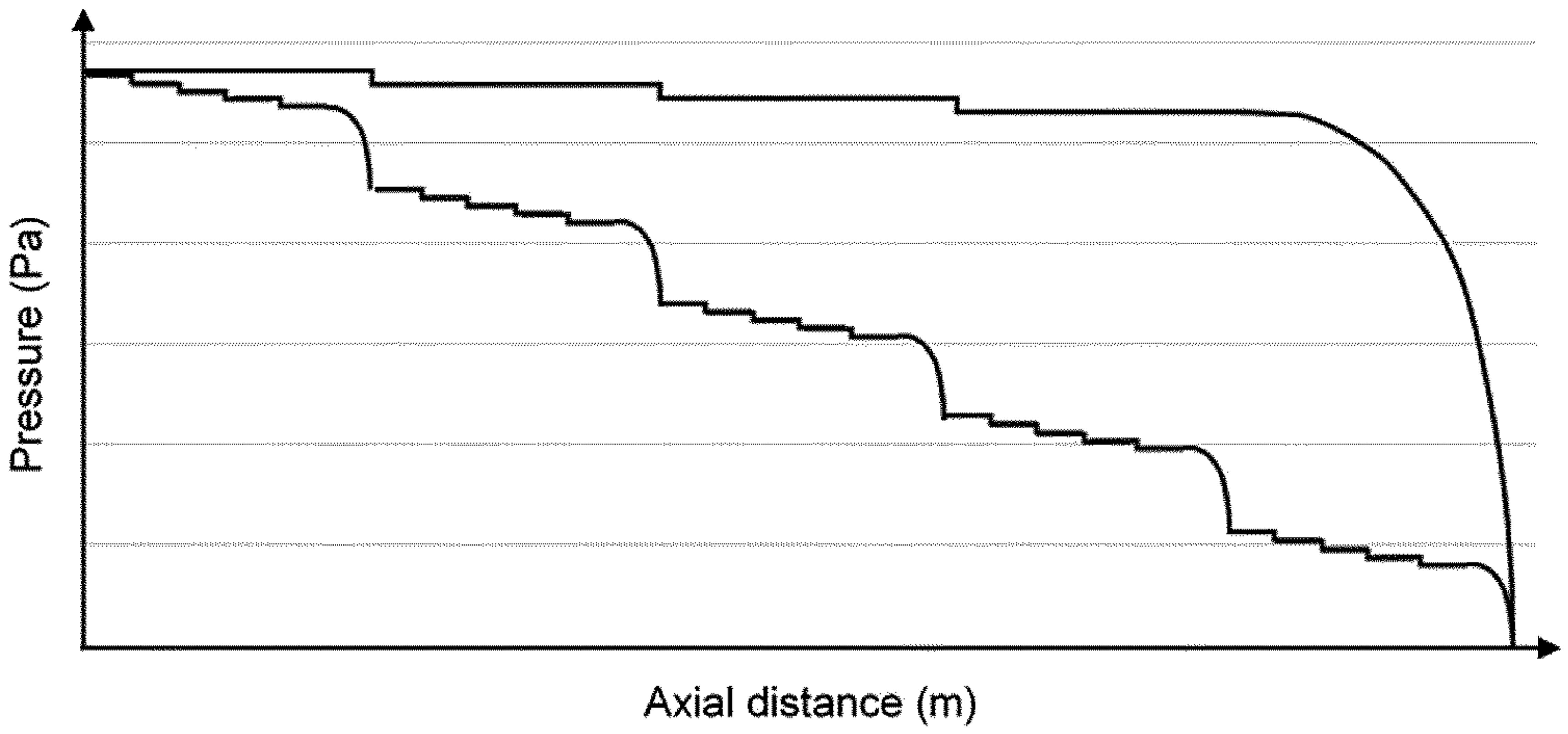
Pressure (Pa)
Axial distance (m)

Fig. 8

INTERNATIONAL SEARCH REPORT

International application No
PCT/EP2020/084234

A. CLASSIFICATION OF SUBJECT MATTER
INV. F01D1/26 F01D1/24 F01D1/32 F01D15/06
ADD.

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
F01D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

EPO-Internal, WPI Data

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y A	US 2019/218913 A1 (SEN ARNAB [IN] ET AL) 18 July 2019 (2019-07-18) paragraphs [0025], [0028], [0035]; figures 2, 4C, 5, 6	1-4,6,9, 11-17 5,7,8,10
Y A	EP 3 460 199 A1 (GEN ELECTRIC [US]) 27 March 2019 (2019-03-27) paragraph [0031]; figure 2	1-4,6,9, 11-17 5,7,8,10
A	US 2018/230805 A1 (MILLER BRANDON WAYNE [US] ET AL) 16 August 2018 (2018-08-16) figure 2	1-17

☐ Further documents are listed in the continuation of Box C.

☒ See patent family annex.

* Special categories of cited documents :

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

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"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search 27 January 2021	Date of mailing of the international search report 05/02/2021
Name and mailing address of the ISA/ European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Fax: (+31-70) 340-3016	Authorized officer Klados, Iason

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INTERNATIONAL SEARCH REPORT

Information on patent family members

International application No

PCT/EP2020/084234

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