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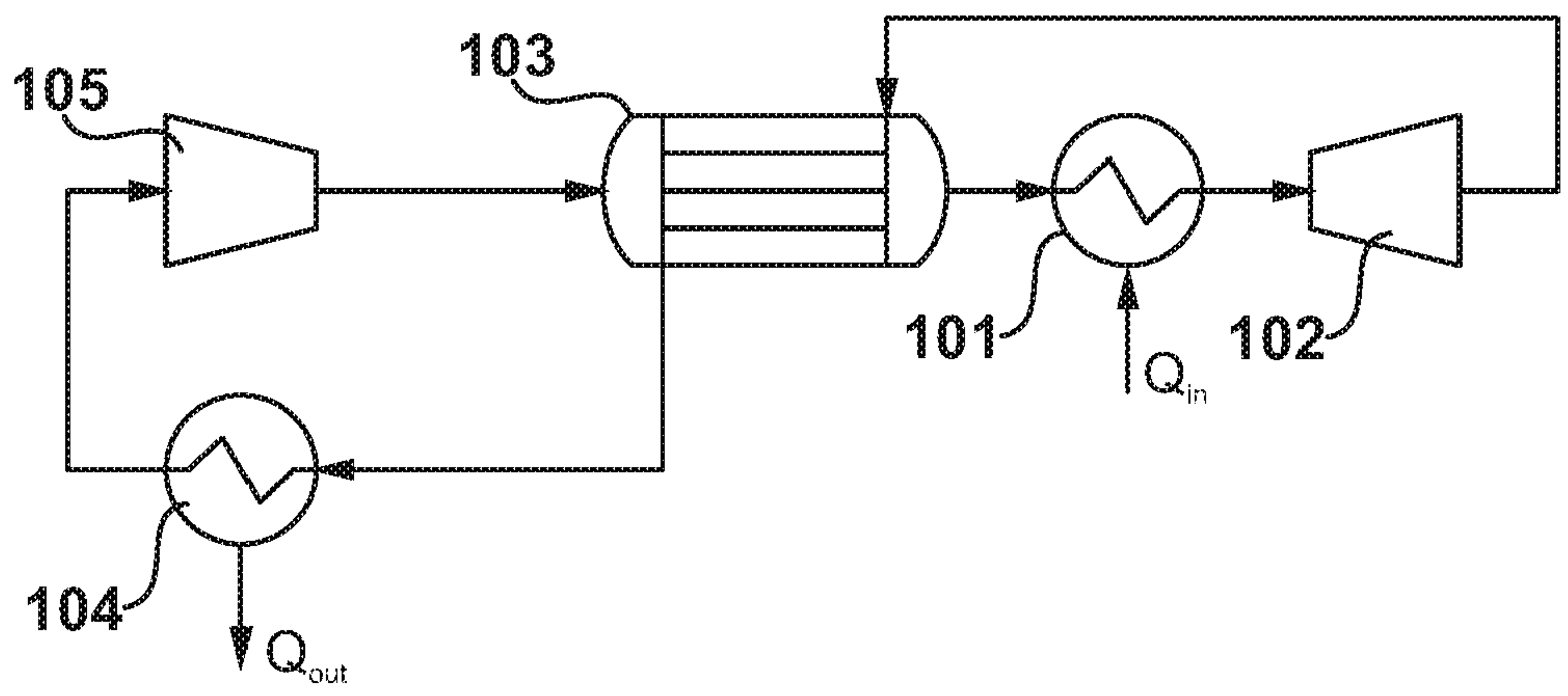

Fig. 1A

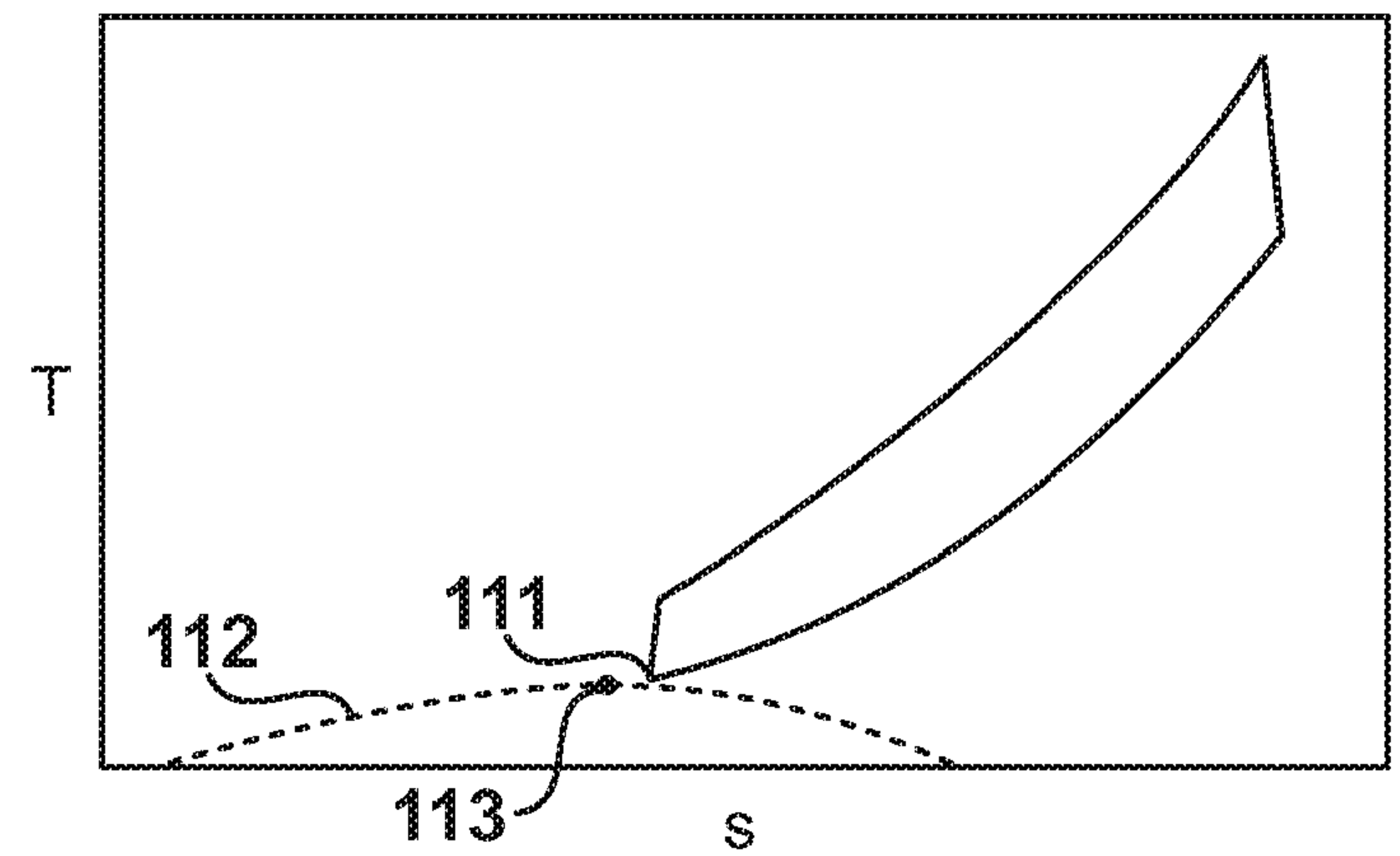

Fig. 1B

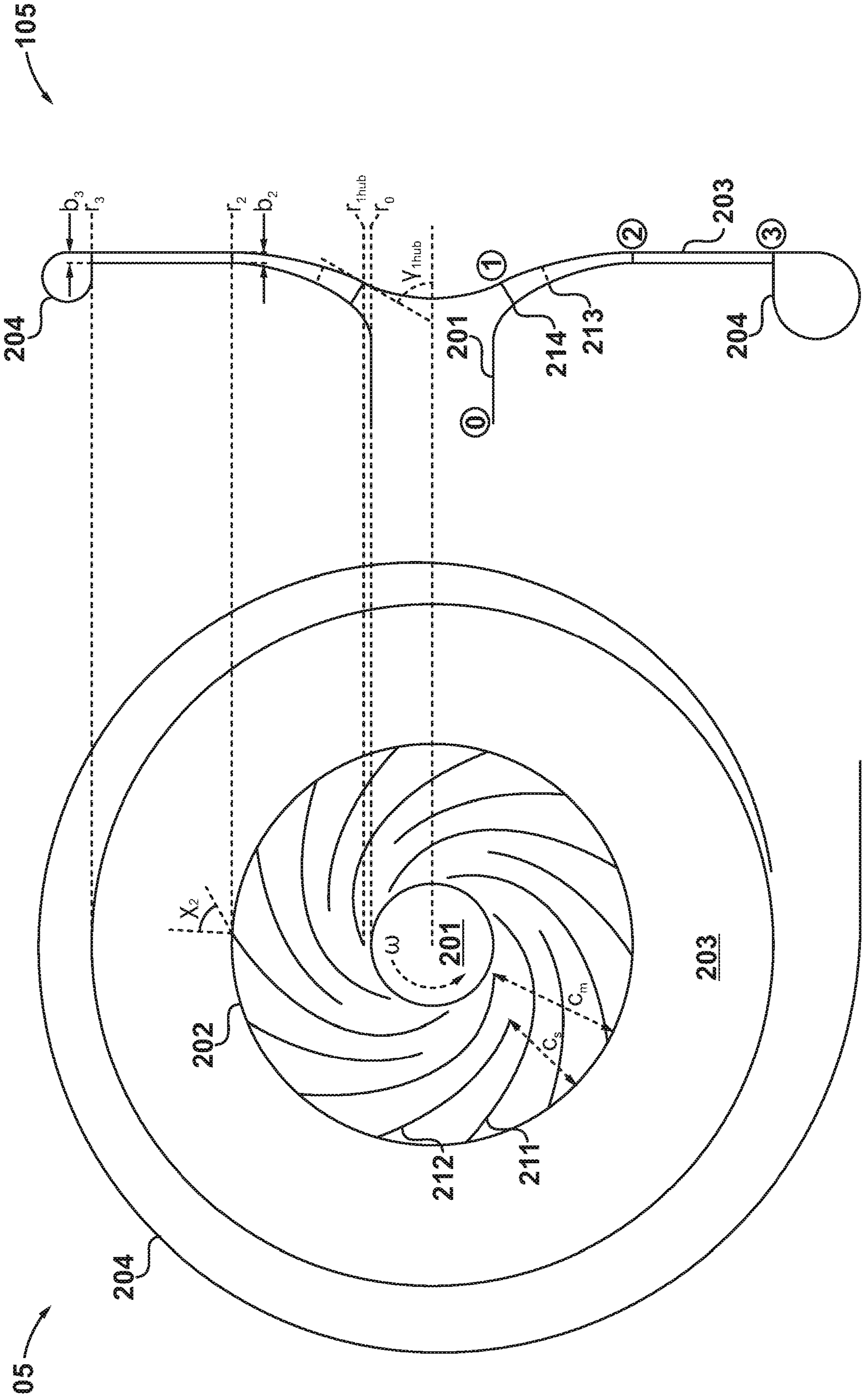
105
204
b_3
r_3
r_2
b_2
r_{1hub}
r_0
γ_{1hub}
201
214
213
0
1
2
3
203
204
202
χ_2
ω
201
203
c_m
c_s
212
211
204
105

Fig. 2B

Fig. 2A

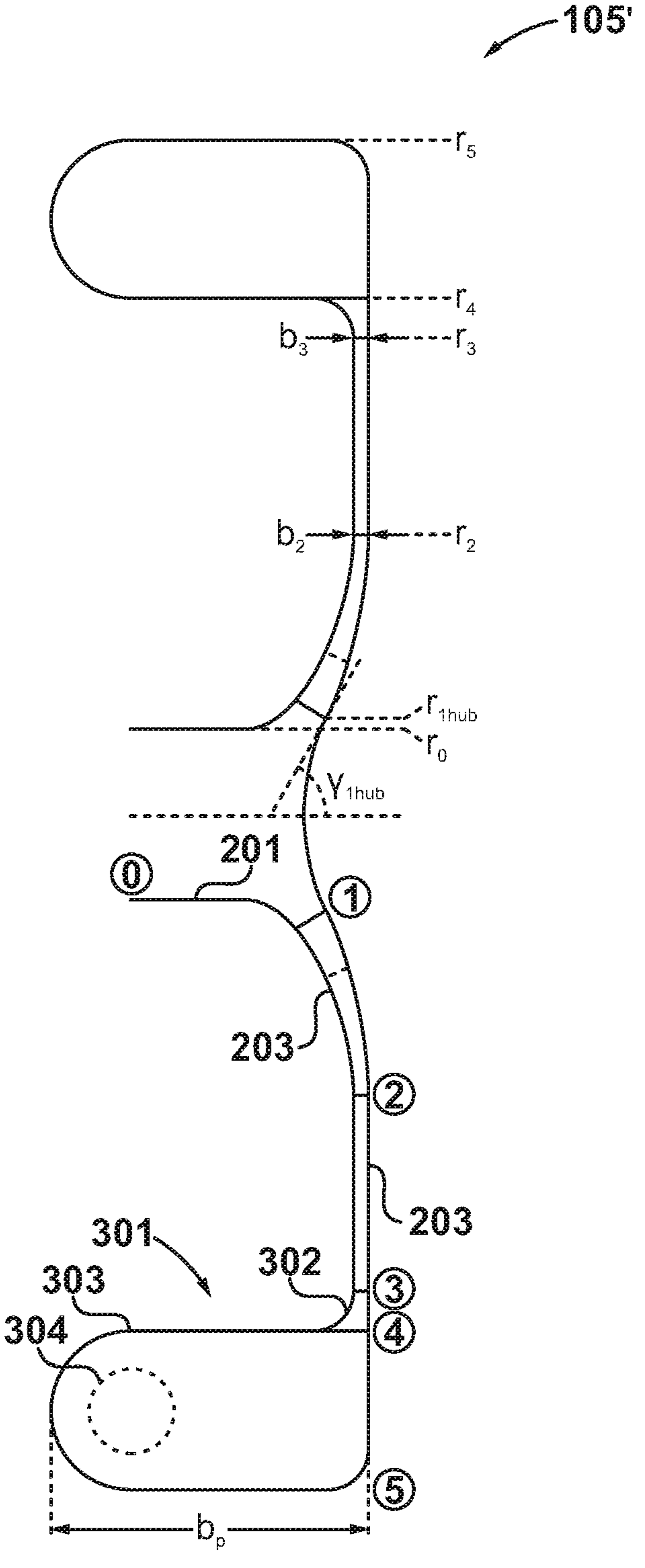
105'
r5
r4
b3
r3
b2
r2
r1hub
r0
γ1hub
0
201
1
203
2
203
301
302
303
3
4
304
5
bp

Fig. 3

SUPERCRITICAL CARBON DIOXIDE COMPRESSOR

TECHNICAL FIELD

This disclosure relates to turbomachinery for compressing supercritical carbon dioxide.

BACKGROUND

Whilst the majority of electric power generation using thermal cycles are either open, direct-heated Brayton cycles such as air-breathing gas turbines, or closed, indirect-heated Rankine cycles such as steam turbines, advances in materials technology have made the use of more exotic working fluids feasible.

One working fluid that shows promise for increased efficiency is carbon dioxide (CO_2), which may be used in a closed, indirect-heated Brayton cycle. CO_2 is attractive because, whilst it becomes supercritical at a fairly high pressure of 7.39 megapascals, its critical temperature is fairly low at 304.25 kelvin which means that heat may be rejected from the cycle at close to ambient temperatures. Further, in its supercritical state, CO_2 has an extremely high density (468 kilograms per cubic metre at the critical point), which reduces the attendant size of the turbomachinery used in the cycle.

Whilst there is dense literature on cycle design, little work has been done to investigate and propose practical implementations of turbomachinery that is suitable for compressing supercritical CO_2 (hereinafter sCO_2).

For example, it is desirable to operate the turbomachine with inlet conditions close to the critical point, as this enables a high pressure rise per unit work. However, doing so means that even small perturbations in inlet conditions can result in the compressibility factor Z of the fluid changing rapidly to be more gas-like than liquid-like. As the fluid becomes more compressible, the working line of the compressor moves as more work is required to achieve a given pressure rise. Unstable operation may therefore ensue if the working line moves too suddenly or too much.

SUMMARY

The invention is directed to turbomachinery suitable for compressing supercritical carbon dioxide, and methods of operation thereof.

In an aspect, a turbomachine of the aforesaid type is provided, the turbomachine comprising, in fluid flow series:

an inlet;

an inducerless radial impeller having a plurality of blades; and

a fully vaneless diffuser;

wherein the radius of the inlet (r_0) is from 25 to 50 percent of the radius of the impeller (r_2).

BRIEF DESCRIPTION OF THE DRAWINGS

Embodiments will now be described by way of example only with reference to the accompanying drawings, in which:

Figure 1A is a schematic of a recuperated sCO_2 Brayton cycle, including turbomachinery to compress and expand a CO_2 working fluid;

Figure 1B is a temperature-entropy (T–s) diagram of the cycle of Figure 1A;

Figure 2A is a plan view of the compressor of Figure 1A;

Figure 2B shows the annulus lines of the compressor of Figure 1A; and

Figure 3 shows an alternative configuration of the compressor.

DETAILED DESCRIPTION

A schematic of a recuperated sCO_2 Brayton cycle is shown in Figure 1A.

The cycle comprises a heater in the form of a first heat exchanger 101, which adds heat, Q_{in}, to the CO_2 working fluid. The heat may be waste heat from another cycle, with the cycle of Figure 1A acting as a bottoming cycle, or from any other heat source such as a solar thermal collector, for example.

The heated working fluid is then expanded through a first turbomachine suitable therefor in the form of a turbine 102 to develop shaft power. Following expansion, the CO_2 working fluid from the turbine 102 is passed through a recuperator 103 to reduce its temperature. Heat is rejected from the cycle, Q_{out}, by a cooler in the form of a second heat exchanger 104.

The cooled CO_2 working fluid is then compressed by a second turbomachine suitable therefor, in the form of a compressor 105. Following the compression stage, a quantity of heat is added in the recuperator 103 and the fluid returns to the first heat exchanger 101 for further heating.

Figure 1B shows a T–s diagram of the cycle of Figure 1A. The heat rejection in the cooler 104 reduces the temperature of the CO_2 working fluid to a minimum point 111 close to the saturation line 112 and the critical point 113 thereon. In this way, the pressure of the working fluid may be increased whilst incurring a minimal increase in temperature.

However, as will be appreciated by those skilled in the art, it is in this region that the properties of the CO_2 working fluid are liable to change rapidly.

First, the speed of sound in the CO_2 drops to 30 metres per second at the critical point. At constant entropy, it rises to over 120 metres per second with only a 0.1 kelvin temperature increase. This leads to the possibility of high Mach number flow when operating turbomachinery near the critical point.

Second, as the CO_2 working fluid enters the compressor 105, it is possible for it to drop in a thermodynamic sense below the saturation line. It is still unknown as to whether a CO_2 working fluid will, in a cycle of the type shown in Figure 1A, actually condense, and, even if it does, what effect this will have.

Thus the embodiments of the compressor 105 described herein provide a turbomachine suitable for compressing sCO_2 that take into account these phenomena. Figures 2A and 2B show the compressor 105 in plan view and meridional cross section respectively.

The compressor 105 comprises, in fluid flow series, an inlet 201 between stations 0 and 1, an impeller 202 between stations 1 and 2, a diffuser 203 between stations 2 and 3, and, in the present embodiment, a volute 204 following the diffuser 203.

In the present embodiment, the compressor 105 has design inlet conditions of 306 kelvin and at 7.7 megapascals, i.e. just above the critical point of the CO_2 working fluid. Further, the compressor 105 is configured to have a design point stagnation pressure ratio of 2.

As described previously, the properties of the CO_2 working fluid around the critical point impose a requirement for stable operation of the turbomachinery across a wide range of conditions.

Thus, the impeller 202 is inducerless, i.e. it does not include an initial set of blades configured to create an axial pressure rise. Instead, the impeller 202 is a purely radial impeller, configured to produce only a centrifugal pressure rise in the CO_2 working fluid. This reduces any time period in which the flow is subcritical, which may occur as the fluid accelerates through the impeller. Further, the radial impeller will continue to operate stably with little or no pressure drop should it enter stall.

To reduce the risk of condensation, the inlet 201 is large relative to the size of the impeller to facilitate sufficient margin in inlet velocity for a given mass flow to the velocity at which the flow becomes subcritical. The radius of the inlet r_0 is thus from 25 to 50 percent of the radius of the impeller r_2.

The radius of the inlet r_0 may alternatively be from 30 to 50 percent of the radius of the impeller r_2. In the specific embodiment of Figures 2A and 2B, r_0 is 34 percent of r_2.

Further, the diffuser 203 is fully vaneless, i.e. there is no vaned space in addition to vaneless space. This provides the widest possible operating range due to increased stability margin. (Vaneless diffusers are less susceptible to stall under low flow conditions than vaned diffusers.)

In the embodiment shown in Figures 2A and 2B, the inlet 201 is radially flared so as to introduce a radial component in the flow prior to entry into the impeller 202 at station 2. In an embodiment, this may be achieved by configuring the annulus lines of the compressor 105 (Figure 2B) so that the hub hade angle at station 1, denoted γ_{1hub}, to between 50 and 70 degrees. In the present example, the hub hade angle γ_{1hub} is 60 degrees to strike a balance between amount of flow turning and reducing risk of separation.

To further increase stability margin, in the present embodiment the impeller 202 has backswept blades. Compressors typically feature only modest backsweep to keep tip speeds and peak stresses under control. However, the use of CO_2 as the working fluid and its attendant high density results in a lower impeller tip radius for a given shaft speed than the equivalent air compressor operating at the same pressure ratio. Consequently, centrifugal loading is reduced. In terms of stress, the impeller 202 experiences, like a pump, predominantly blade pressure forces. These are dictated primarily by blade height, rather than backsweep.

Thus an opportunity exists to implement high levels of backsweep, which reduces the absolute Mach number at the impeller exit thereby reducing losses in the diffuser 203 and improving efficiency. In the present embodiment, for instance, the flow relative Mach number at the entry to the diffuser 203 is 0.44.

The sweep of a blade in a radial compressor may be defined by the blade exit angle, which is also known as blade metal angle. This angle is denoted χ_2, distinguishing it from the relative exit flow angle β_2, and is defined relative to the radial direction at the blade tip. The sign convention for χ_2 is such that positive values denote forward sweep, i.e. in the intended direction of rotation ω, whilst negative values denote negative sweep, as is the case with impeller 202. In the present embodiments, χ_2 may for example from –50 to –70 degrees. In the specific embodiment of Figures 2A and 2B, χ_2 is –60 degrees.

The use of backsweep also increases the degree of reaction Λ of the compressor 105, i.e. the enthalpy rise in the rotor as a proportion of the whole stage. This is beneficial as it is more challenging to achieve high pressure rise in the diffuser 203.

In the present embodiment, the number of blades in the impeller satisfies the requirement that the velocity difference between the suction and pressure surfaces thereon is less than twice the meanline velocity. Thus, in the specific embodiment shown in Figure 2A the impeller 202 has 14 blades in total. It will be appreciated, however, that in other implementations the blade count may differ.

In the present embodiment, the impeller 202 has a set of main blades 211 and a set of splitter blades 212. In the specific embodiment shown in Figure 2A, there is an even number main and splitter blades – one splitter blade for every main blade. The splitters are provided such that the impeller 202 is not under-bladed at the exit radius r_2, which ensuring that there is not an excess of blades at the inlet radius r_1 which would act to increase blockage and the likelihood of condensation.

In the present example, each splitter blade 212 has a leading edge 213 located 30 percent of meridional chord from the leading edge 214 of each main blade 211. Thus the meridional chord length of the splitter blades 212, denoted c_s, is 70 percent of the meridional chord length of the splitter blades 212, denoted c_m. Each splitter blade 212 is located in the middle of the passage formed between adjacent main blades 211.

As described previously, the diffuser 203 is a fully-vaneless diffuser. Whilst vaned diffusers may give higher efficiencies at their design point, they exhibit reduced stability off-design due to flow separation. A fully vaneless diffuser therefore provides a wider operating range.

In the present embodiment, the length of the diffuser 203 satisfies a requirement to maximise pressure recovery whilst minimising viscous losses. Thus, in an embodiment the radius at the diffuser exit, r_3, is from 1.2 to 1.8 times greater than the radius at the diffuser entry, r_2. The radius at the diffuser exit, r_3, may in another embodiment be from 1.3 to 1.7 times greater than the radius at the diffuser entry, r_2. In the specific embodiment shown in Figure 2A, r_3 is 1.7 times greater than r_2.

Pressure recovery is aided by, in the present embodiment, having a non-varying passage height for the diffuser 203 over its radial extent, i.e. the height of the diffuser passage at its entry, b_2, is the same as the height of the diffuser passage at its exit, b_3. The diffuser 203 therefore has an annulus height ratio b_3/b_2 of 1.

The volute 204 in the specific embodiment shown in Figure 2B is of asymmetric configuration, but it will be appreciated that a symmetrical configuration may be used instead.

In the present embodiment, the flow area A of the volute at the tongue is equal to the flow area at the exit of the diffuser 203. This prevents diffusion and thus avoids static pressure distortion at the exit of the diffuser 203, which may affect the stability of the compressor 105.

In operation as part of the cycle of Figure 1A, the compressor 105 is provided with a supply of sCO_2, and the impeller 202 is rotated by the turbine 102. For design point operation, the sCO_2 may be provided at the conditions discussed above of 306 kelvin and at 7.7 megapascals, and the impeller 102 may be rotated at 50000 revolutions per minute to achieve the design stagnation pressure ratio of 2. Off-design operation may, however, still be carried out reliably due to the combination of measures discussed herein to improve stability.

For example, the impeller may be rotated at a speed greater than 50000 revolutions per minute, such as 70000 revolutions per minute, to achieve a stagnation pressure ratio of from 3 to 4.

Alternatively a speed less than 50000 revolutions per minute may be used to achieve a stagnation pressure ratio of from 1 to 2.

Alternatively, the compressor may be provided with a different design point parameter set depending on the overall cycle requirements.

A different embodiment of the compressor is shown in Figure 3, and is identified as compressor 105'. The compressor 105' is largely the same in configuration as compressor 105, and thus includes inlet 201, impeller 202 and diffuser 203 as described previously. However, in the embodiment shown in the Figure, the collector is a plenum 301 rather than the volute 205.

In the present example, the plenum 301 is substantially axisymmetric. This may simplify manufacture. The plenum 301 includes an offtake 302 for connection of the compressor 105' to the rest of the cycle of Figure 1. In the present embodiment, the plenum 301 includes an entrance 302 extending radially between station 3 and a station 4, and a chamber 303 between station 4 and a station 5. In the present example, the radial extent of the chamber 303, equal to $r_5 – r_4$, is one third of the length of the diffuser 203, i.e. $r_3 = 3(r_5 – r_4)$. In the present embodiment, the chamber 303 has a height that is at least twice its width.

As shown in Figure 3, the configuration of the plenum 301 is such that the offtake 302 is located out of the plane of the diffuser 203. This may assist in terms of reducing swirl in the flow exiting the diffuser 203, particularly as it is of a fully vaneless type. Whilst the total amount of swirl may be reduced, there may still be a small component and thus in the present embodiment the offtake 304 is oriented tangentially with respect to the flow direction to minimise losses.

In the present embodiment, the offtake 302 has a cross-sectional area equal to the cross-sectional area of the inlet 201 divided by the design stagnation point pressure ratio. In the present example therefore, in which the compressor 105' has a design point stagnation pressure ratio of 2, the offtake 302 has a cross sectional area that is half that of the intake 201. Thus, as the design point, the flow rate into and out of the compressor 105' may be substantially equal.

CLAIMS

1. A turbomachine configured to compress supercritical carbon dioxide, the turbomachine comprising, in fluid flow series:

an inlet;

an inducerless radial impeller having a plurality of blades; and

a fully vaneless diffuser;

wherein the radius of the inlet (r_0) is from 25 to 50 percent of the radius of the impeller (r_2) and

each of the plurality of blades is a backswept blade, having a blade exit angle (χ_2) of from –50 to –70 degrees.

2. The turbomachine of claim 1, in which the radius of the inlet (r_0) is from 30 to 50 percent of the radius of the impeller (r_2).

3. The turbomachine of claim 1, in which each of the plurality of blades have a blade exit angle (χ_2) of –60 degrees.

4. The turbomachine of claim 1, in which the plurality of blades comprises:

a set of main blades; and

a set of splitter blades.

5. The turbomachine of claim 4, in which a meridional chord length of the splitter blades (c_s) is 70 percent of a meridional chord length of the main blades (c_m).

6. The turbomachine of claim 4, in which the impeller comprises one splitter blade for each main blade.

7. The turbomachine of claim 1, in which the diffuser has an annulus height ratio (b_3/b_2) of 1.

8. The turbomachine of claim 1, in which the radius of the diffuser (r_3) is from 1.2 to 1.8 times larger than the radius of the impeller (r_2).

9. The turbomachine of claim 8, in which the radius of the diffuser (r_3) is from 1.3 to 1.7 times larger than the radius of the impeller (r_2).

10. The turbomachine of claim 1, having a design point stagnation pressure ratio of 2 or greater.

11. The turbomachine of claim 1, further comprising a plenum arranged to receive fluid from the diffuser, said plenum comprising an offtake having a cross-sectional area equal to the cross-sectional area of the inlet divided by the design point stagnation pressure ratio of the turbomachine.

12. A method of operating the turbomachine of claim 1, comprising:

supplying supercritical carbon dioxide to the inlet of the turbomachine; and

rotating the impeller.

13. The method of claim 12, in which the supercritical carbon dioxide is supplied at 306 kelvin and at 7.7 megapascals.

14. The method of claim 13, in which the impeller is rotated at 50000 revolutions per minute or more.

15. A closed, indirect-heated Brayton cycle having a carbon dioxide working fluid and comprising the turbomachine of claim 1.