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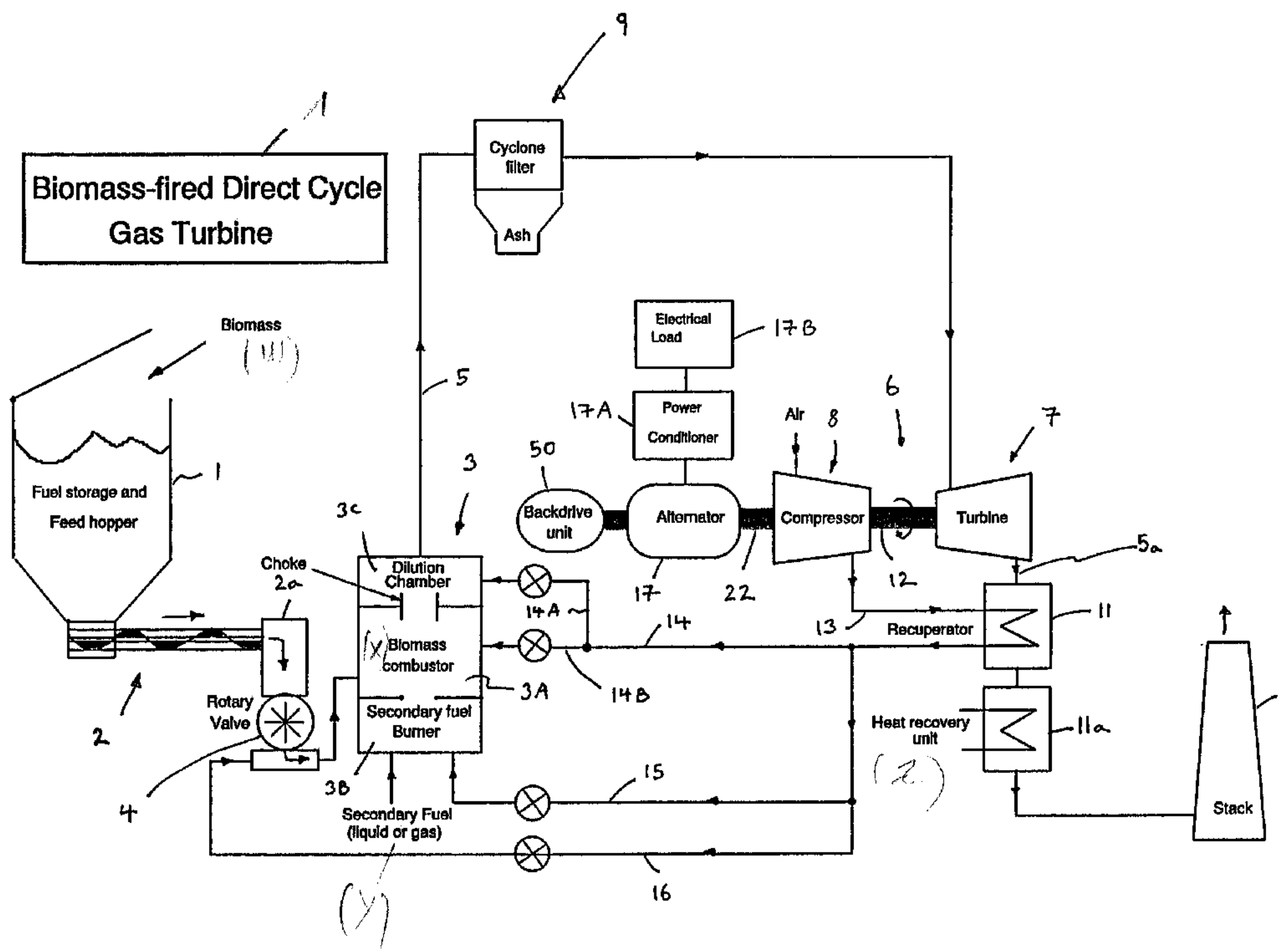

(57) Abrégé/Abstract:
A biomass fuel combustor for use in the pressurized combustion comprising of biomass fule particles to produce a pressurized exhaust gas, the combustor comprising: a cyclonic combustion chamber having a combustion region and first and second fuel inlets, the first inlet being for the entry into the chamber of gas and/or liquid secondary fuel for combustion in said combustion region and the second inlet being for the entry into the chamber of biomass fuel particles also for combustion in said combustion region, the chamber being so constructed and arranged that the heat generated by the secondary fuel combustion will, in use and at least during start-up of the combustor, promote fragmentation of the incoming biomass fuel particles.

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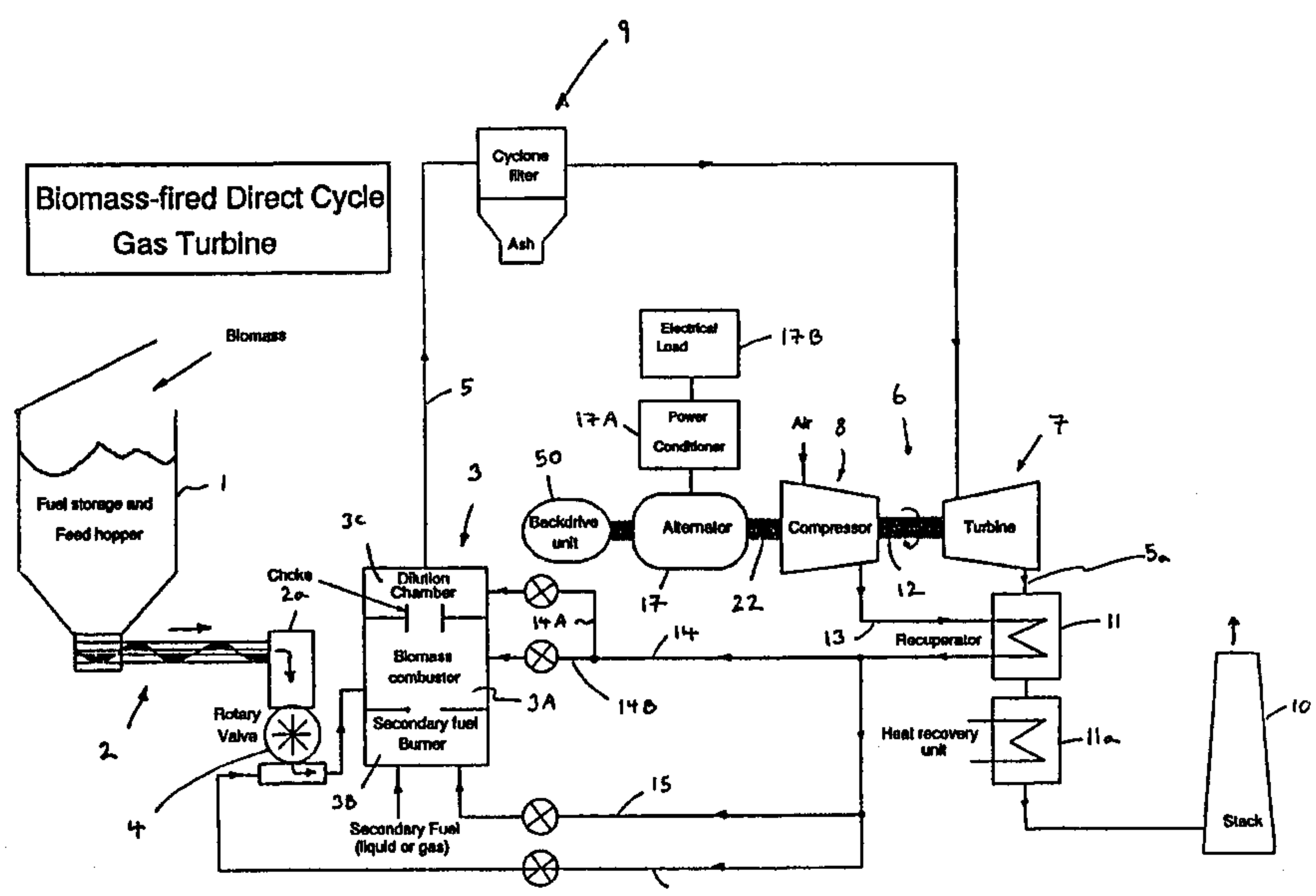


(57) Abstract

A biomass fuel combustor for use in the pressurized combustion comprising of biomass fule particles to produce a pressurized exhaust gas, the combustor comprising: a cyclonic combustion chamber having a combustion region and first and second fuel inlets, the first inlet being for the entry into the chamber of gas and/or liquid secondary fuel for combustion in said combustion region and the second inlet being for the entry into the chamber of biomass fuel particles also for combustion in said combustion region, the chamber being so constructed and arranged that the heat generated by the secondary fuel combustion will, in use and at least during start–up of the combustor, promote fragmentation of the incoming biomass fuel particles.

BIOMASS COMBUSTION CHAMBER FOR GAS TURBINE

This invention relates to biomass fuelled gas turbines, particularly but not exclusively to a direct cycle biomass-fired gas turbine system, a biomass fuel combustor suitable for use in such a system and a process for the pressurized combustion of biomass fuel to produce power using a gas turbine system.

Biomass is a source of renewable energy that is "carbon dioxide neutral", i.e. it does not contribute to the global greenhouse effect. Utilization of biomass for the generation of electricity is, therefore, of considerable potential benefit to the environment but is, at present, uneconomical. Combined heat and power (CHP) units offer high fuel conversion efficiencies (fuel utilization factors in excess of 75%) provided that the "heat component" is of sufficiently high grade. This requirement is met most satisfactorily by the gas turbine.

The simplest, and most cost-effective, way to utilize biomass fuel in a gas turbine is to burn it in the gas turbine's combustion chamber and pass the combustion gases directly through the turbine blades (this is what is understood by "direct cycle"). Unfortunately, the combustion of biomass leads to the formation of chemical compounds which, if allowed to deposit on turbine blade surfaces, can cause severe deterioration of performance by blockage, erosion and corrosion. Alternative methods of attempting to use biomass to fuel a gas turbine, involving pyrolysis, gasification and isolation of the combustion gases from the turbine by large heat exchangers, all give rise to high biomass conversion costs.

High performance gas turbines specifically developed for aerospace and land based applications have been utilized in direct cycle biomass-fired plants. Not only are such gas turbines expensive, accounting for up to half of the capital cost of the whole CHP plant, but their high inlet temperatures (> 1000°C) and pressure ratios (> 10:1), necessary for high efficiency propulsion and shaft power applications, render them unsuitable for biomass conversion where temperatures are generally restricted to less than 830°C in order to minimise the problems of deposition, erosion and corrosion (DEC) and pressure ratios optimize at about 5:1 for maximum efficiency in a recuperated cycle.

Aerospace-developed gas turbine engines also present operational problems when coupled to relatively massive biomass combustion systems (for example, ones with a combustor volume of $>1m^3$), in that their dynamic characteristics are better suited to small, rapid response combustors.

High performance industrial (land-based) gas turbines are generally able to cope with larger combustion systems, but still suffer from the same problems of requiring high turbine inlet temperatures and pressure ratios for efficient operation with maximum electrical output or shaft power.

According to a first aspect of the present invention there is provided a biomass fuel combustor for use in the pressurized combustion of biomass fuel particles to produce a pressurized exhaust gas, the combustor comprising:

a cyclonic combustion chamber having a combustion region and first and second fuel inlets, the first inlet being for the entry into the chamber of gas and/or liquid secondary fuel for combustion in said combustion region and the second inlet being for the entry into the chamber of biomass fuel particles also for combustion in said combustion region, the chamber being so constructed and arranged that the heat generated by the secondary fuel combustion will, in use and at least during start-up of the combustor, promote fragmentation of the incoming biomass fuel particles.

The smaller that a gas turbine combustor can be made, the less expensive it is likely to be and the more rapidly it will be able to respond to operational demands that are imposed upon it, e.g. sudden changes in electrical load. It is important, however, that the combustor is not so small that the fuel is incompletely burned, otherwise efficiency and environmental emissions will suffer.

The minimum size of a combustor is primarily determined by the rate at which heat is released from the fuel, i.e. how fast if can be burned, safely and efficiently.

Provided that the fuel and oxygen (air) are well-mixed, the combustion rate is largely dependent on the rate of evaporation or devolatilisation of fuel particles which is, in turn, dependent on their surface-to-volume ratios.

With liquid fuels it is comparatively easy to produce small particles (droplets of diameter typically 10 to 100 micron) by "atomising" the liquid by forcing it though small orifices (or jets). With wood it is possible to produce small particles by progressively grinding larger particles but this process of comminution requires specialised (expensive) machinery and is wasteful of energy.

The method hereinafter described depends upon the disintegration of a wood (or biomass) cell that occurs when it is rapidly heated. Certain constituents of the cell interior change phase from a gel to a vapour, with such rapid increase in volume that the cell wall membrane is unable to contain the pressure thus generated and ruptures with explosive force scattering the millions of cellulose fibres comprising the membrane. These fibres, which would otherwise coalesce to form a slow burning char, are free to react rapidly with the air into which they are dispersed. The hiss and crackle when grass or brushwood is thrown onto a very hot bonfire is indicative of the acoustic energy released by these explosive forces.

A second, though no less important, feature of this mechanism is that the resultant ash particles are also very small (sub-micron in diameter) having high drag-to-weight coefficients which cause them to be entrained by the gas flow. If the gas is then cooled rapidly to a temperature that prevents or impedes particle coalescence and growth, most (e.g. > 95%) of the ash particles can be made to pass between the turbine blades without contacting the surface of the blades. Thus DEC is prevented.

According to a second aspect of the present invention there is provided a direct cycle biomass-fired gas turbine system, comprising:

a biomass combustor for the pressurized combustion therein of biomass; and

a gas turbine including a turbine section, arranged to receive and be driven by the hot pressurized combustion gas resulting from said biomass combustion, and a compressor section,

wherein the gas turbine is so constructed and arranged that, when operated in a recuperated cycle at a turbine section gas inlet pressure in the range of 3-7 times atmospheric pressure, the gas turbine will work at maximum efficiency.

By being a direct cycle the complication and cost of pyrolising or gasifying the biomass, or seeking to isolate the biomass fuel combustion gases from the turbine by large heat exchangers, can be avoided.

Advantageously, the combustor in the system according to the second aspect of the present invention is the combustor of the first aspect of the present invention.

According to a third aspect of the present invention there is provided a process for the pressurized combustion of biomass fuel to produce power, the method comprising: providing a system in accordance with the second aspect of the present invention;

combusting biomass fuel under pressure in the combustor to produce hot pressurized combustion gas;

expanding said hot pressurized combustion gas through the turbine section of the gas turbine to produce a power output.

The gas turbine is specifically designed to cope with the special requirements of biomass combustion. Its performance is optimised at turbine inlet temperatures below 900°C, for example around 850°C. When recuperated it has a maximum efficiency at pressure ratios in the range 3:1 to 7:1, for example around 5:1. It is also capable of coping with limited amounts of DEC, and is capable of handling large, slow response biomass combustors, for example ones of > 1m^3 internal volume.

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

Figure 1 is a schematic of an embodiment of a biomass-fired direct cycle gas turbine system;

Figure 2 illustrates an embodiment of a gas turbine suitable for use in the system of Figure 1;

Figure 3 is a cross-sectional view of an alternative embodiment of a basic turbine (with ancillaries and pipework removed) suitable for use in the system of Figure 1;

Figure 4 illustrates, in longitudinal cross-section, an embodiment of combustor suitable for use in the system of Figure 1; and

Figure 5 illustrates, in longitudinal cross-section, another embodiment of combustor suitable for use in the system of Figure 1.

Figure 1 illustrates schematically an embodiment of biomass-fired direct cycle gas turbine system.

So as to minimise problems with deposition, erosion and corrosion (DEC) of the gas turbine, the preferred biomass fuel for the system of the present invention is wood. The reason for this is that many woods have a relatively low ash content (less than 2%), as well as a high ash fusion temperature, for example greater than 800°C. In contrast, coal has an ash content of the order of 15-30%. In addition to there being less of it, the ash produced by burning wood tends to be less abrasive than that produced by burning coal. The wood chips preferably have a particle size of maximum dimension up to about 20mm.

In the illustrated embodiment of system, biomass fuel is stored in a feed hopper 1. At the base of the hopper a screw auger 2 (such as a modified Veto unit produced by Ala-talkari) is provided to feed the biomass fuel continuously from the hopper 1, through the intermediate hopper 2a, to the biomass combustor 3 via one or more rotary valves 4, (such as a Blowing Seal produced by Rotolok). The inlet end of the auger 2 is at approximately atmospheric pressure and the interior of the biomass combustor 3 has an operating pressure of typically 3-5 barg (as will be explained later). The comparatively low pressure ratio inside the combustor 3 ensures that sufficient pressure sealing is achieved by the rotary valve(s) 4 combined with the flow impedance of the auger 2 to avoid significant loss (> 1%) of combustor pressure.

Although the biomass combustor 3 will be described in more detail later in conjunction with Figures 3 and 4, in describing the system of Figure 1 a brief description of the combustor will be included.

In Figure 1 the biomass combustor is shown schematically as including a central combustion chamber 3A. At the lower end of the combustor 3 in Figure 1 there is shown a secondary liquid/gas fuel burner 3B, whose purpose will be described later. At the upper end of the combustor 3 there is shown a dilution chamber 3C. The purpose of this chamber 3C will also be described later. Depending on the detailed combustor design, it may be mounted with its axis horizontal or vertical.

For present purposes it is sufficient to say that, in use, hot pressurized combustion gas from biomass and/or secondary fuel combustion exits the combustor 3 along exhaust pipe 5.

The system includes a gas turbine 6, comprising a turbine section 7 and a compressor section 8. Before reaching the inlet to the turbine 7 the hot pressurized combustion gas passing along exhaust pipe 5 may be fed through a filter 9 typically a cyclone filter, to remove any of the larger ash particles or other debris carried in the gas. Whilst this optional cyclone filter 9 may not remove very small ash particles of typically less than 1 micron, the particles that it cannot remove are generally of a sufficiently small size as to cause minimal deposition, erosion and corrosion problems to the turbine stage of the turbine section 7 when passing therethrough.

Upon entering the inlet to the turbine section 7 the hot pressurized combustion gas is expanded through the turbine and exits the turbine section 7 via a further exhaust pipe 5a. In the illustrated embodiment these waste exhaust gases are vented via a stack 10. Prior to being vented, the waste exhaust gases passing through pipe 5a (as shown in Figure 1) pass through a heat exchanger or recuperator 11 to exchange heat with compressed air exiting the gas turbine's compressor section 8. Thereafter the exhaust gases may, as shown, pass through a heat recovery unit 11a which will transfer heat as part of an overall CHP plant.

As will be explained in more detail in conjunction with Figures 2 and 3 below, the turbine and compressor sections 7,8 of the gas turbine 6 are connected by a common shaft 12, such that driving of the turbine section 7 by the expansion therethrough of the hot pressurized combustor gas causes the compressor section 8 to intake air and to output compressed air via a pipe 13. In the Figure 1 illustrated embodiment this compressed air in pipe 13 is passed through the heat exchanger or recuperator 11 prior to the pipe splitting into several feeds 14,15,16.

Compressed air feed 14 itself splits into two feeds 14A,14B. As will be explained below, feed 14A is routed to the dilution chamber 3C of the combustor 3 and feed 14B is routed to the combustion chamber 3A of the combustor 3.

Feed 15 goes to the secondary fuel burner 3B.

Compressed air feed 16 goes to the rotary valve 4, such that the biomass fuel injected into the combustor 3 via the rotary valve is, in fact, a mixture of biomass fuel particles and compressed air.

Feeds 14A, 14B, 15 and 16 are, as shown schematically , advantageously provided with control valves to regulate air flow.

In order to provide a power output the shaft of the gas turbine 6 is linked to an alternator 17 for the generation of electrical power, this alternator being provided with a power conditioner 17A and an electrical load 17B in a known manner.

A backdrive unit (also known as a variable frequency speed control unit) 50 may, as shown, also be provided in series with the alternator 17 to act like a starter motor to enable rapid start-ups. At start-up the backdrive unit can be used to spin the gas turbine 6 up to self-sustaining speed (with zero load at the alternator).

The construction of the gas turbine will now be looked at in more detail in conjunction with Figures 2 and 3.

The preferred form for the gas turbine 6 is based on a turbocharger adapted from automotive, marine or similar reciprocating engine applications. For this reason the turbine construction will not be described in detail. In contrast to aerospace-developed and high performance industrial land-based gas turbines, a gas turbine of the sort illustrated in Figures 2 and 3 has been found to be very well suited for use with a biomass combustor. In automotive and marine derived turbochargers, the rotating components are generally designed for optimum performance at pressure ratios of the order of 4 to 6, and are also capable (with substitution of higher temperature materials for critical components) of sustaining prolonged operation at temperatures of the order of 850°C. Furthermore, automotive, marine and similar turbochargers are inexpensive relative to high performance gas turbines, and the inventor has found them to be generally suited to incorporation in gas turbines rated from 5 to 500 kWe . The blades of the radial inflow turbine wheel 18 of the turbine

6, as well as its associated stator blades, are configured so as to minimise deposition and impaction of ash particles of the size present in the hot pressurized combustion gas. In this way the turbine 6 can be made far more resistant to performance degradation from deposition, erosion and corrosion than can the turbine section of an axial flow aerospace developed or high performance industrial land-based gas turbine.

In the exemplary process of Figure 1, the hot pressurized combustion gas provided, via pipe 5 to the inlet of the turbine section 7, is at approximately 800°C. After expansion across the radial inflow turbine wheel of the turbine section 7 the combustion gas, now at reduced pressure, exits the turbine section 7 via exhaust pipe 5a.

Although Figure 3 illustrates a slightly different embodiment of gas turbine to that illustrated in Figure 2, the constructions of the Figure 2 and Figure 3 devices are very similar and are conventional for automotive or marine turbochargers. As can most readily be seen in Figure 3, the turbine section 7 contains a single radial inflow turbine wheel 18, constituting a single turbine stage which is conventional for automotive, marine or similar turbochargers.

This turbine wheel 18 is mounted on the opposite end of a bearing-supported shaft 19 to a centrifugal compressor wheel 20, housed in the compressor section 8, constituting a single compressor stage. As in an automotive turbocharger, driven rotation of the centrifugal compressor wheel 20 causes air to be drawn in axially and compressed, prior to its passage to compressor section exit pipe 13. In Figure 2 the compressor inlet is provided with an air filter element 21 to filter the air drawn into the compressor section 8. This filter 21 is omitted in Figure 3.

It will be appreciated that operation of the gas turbine 6 not only produces a supply of compressed air, which exits the turbine compressor section 8 via pipe 13, but drives the alternator 17, either directly or via a gearbox (not shown), to produce electrical power.

The general principles of cyclonic combustors are well understood and only those aspects specific to the present invention are considered here. A suitable background reference is:Syred, N., Claypole, T.C. and MacGregor, S.A., 1987, "Cyclone Combustors", Principles of Combustion Engineering for Boilers, Academic Press.

In the first embodiment of combustor illustrated in Figure 4, a mixture of biomass particles and air is introduced into the combustion chamber 24 via an inlet pipe 25, into an annular region 23 exterior to a choke tube 34 and including a lip or weir 23A. The biomass particles, coming from rotary valve(s) 4, are mixed in a compressed air stream from compressed air feed pipe 16 - see Figure 1. Typical temperatures and pressures for the compressed air stream are 150-300°C and 3-5 barg respectively. A high level of dilution air is fed through pipe 14a, for example approximately twice the amount of air necessary for stoichiometric combustion. This high level of dilution air not only helps to keep the turbine inlet temperature down below the biomass fuel's ash fusion temperature, so as to reduce the likelihood of any ash in the hot pressurized combustion gas sticking to the turbine wheel 18, but it also helps to minimise the size of the ash particles by reducing the likelihood of agglomeration. The smaller the ash particle size the less damage the particles are likely to cause to the turbine wheel 18, and very small particles (typically < 1 micron) can be made to pass between the blades of the turbine without impaction or deposition.

The combustor illustrated in Figure 4 is a cyclonic combustor. To help to encourage tangential flow in the combustion chamber 24, and in the initial combustor region 26, the biomass/air inlet pipe 25 is oriented so that the entering biomass/air mixture includes a significant (generally > 30 m/s) tangential component, relative to the longitudinal axis of the combustor 3. The initial combustion region 26 at the top end of the combustion chamber 24 of the Figure 4 combustor is provided with a further inlet 27. This inlet is part of a secondary fuel burner 3B. In the illustrated embodiment, the entry of the inlet 27 is approximately 180° displaced from the biomass fuel/air inlet pipe 25 and is also oriented so that the flame of the secondary burner 3B also includes a significant (generally > 30 m/s) tangential component. This secondary fuel burner is fuelled by a gas or liquid secondary fuel. The secondary fuel burner, which may for example be a kerosene burner, has a fuel feed 29 and a compressed air feed 15. The auger(s) 2, the compressed air feed 15 and secondary fuel feed 29 can all be regulated by a controller (not shown) according to power demand.

One use of the secondary burner 28 is on start-up of the combustor. Not only can the secondary burner 28 be used to heat up the interior of the combustion chamber 24, it can be used to ignite the initial flow of biomass particles/air entering annular region 23 via feed pipe 25. The high flame temperature of the secondary burner 28 (approximately 1600°C) causes very rapid heating of the incoming biomass particles, causing near instantaneous (less than 10 ms) fragmentation of those particles, accelerating combustion rates. After the biomass has been ignited, the secondary fuel burner can be switched from being completely responsible for combustion in the combustion chamber 24 to being only part responsible. For example, in steady-state conditions the secondary burner 28 may account for up to 10% of the energy input into the combustor. Under these circumstances the rapid heating of the incoming biomass particles will derive from radiative heat transfer from the biomass flame.

Advantageously, the capacity of the secondary burner 28 is such that full load operation of the gas turbine system can be achieved using the secondary burner 28 alone. In this way, any problems with the biomass feed system requiring the system to be shut down need not affect the system's power generation, enabling the biomass feed system to be isolated for maintenance whilst the system as a whole remains on-load.

Whilst it may be possible to reduce the energy input of the secondary burner to less than 10% of the total on a time-average basis, there are benefits in continuing to run the secondary burner. Not only will it then continue to contribute to maintaining the high temperature in the combustion region 26 to cause near instantaneous fragmentation of the biomass particles, it is also very helpful from the point of view of system control. Rapid variations in power demand (faster than 10% in approx 1 second), as well as alternator speed regulation, can be readily satisfied by varying operation of the secondary burner 28.

In contrast to known biomass combustors, in the embodiment of combustor illustrated in Figure 4 the flame can 30 defining the combustion chamber 24 is made of metal (for example stainless steel, or proprietary heat-resistant metals such as the Inconel series of alloys) and is not lined with refractory material. Refractory material in known combustors is present not only to protect the underlying surface forming the combustion chamber against heat damage, but also to provide an extensive radiant non-metallic refractory surface to promote biomass ignition and combustion. The presence of a large amount of refractory material is, however, disadvantageous. Firstly, at start-up from "cold", significant energy is required to be input into the combustor in order to raise the refractory material to a temperature (generally > 1000°C) at which its surface will radiate sufficiently to promote biomass ignition. Secondly, the combustor has large thermal inertia such that it is very difficult to match power generation to rapid and sudden responses in demand. In an extreme situation, a rapid and sudden reduction in demand for power can lead to overspeeding and damage of the associated gas turbine. Thirdly, in known refractory lined combustors any refractory material breaking away from the lining of the combustion chamber can enter the turbine downstream of the combustor causing significant damage to the turbine assembly. Accordingly, the predominantly metal construction of the combustor not only allows fast start-up, but also improves response to transients.

A thin refractory material lining of less than 2 mm thickness may, however, be provided on the interior wall of at least part of the flame can 30.

In the Figure 4 embodiment the flame can 30 is surrounded by a wind box 31 of metal construction, forming a chamber 32 to which compressed air (from the compressor 8) is supplied via compressed air feed 14B. A plurality of air injection openings are provided in the wall of the flame can 30 and these openings are arranged such that pressurized cooling air introduced into the chamber 32, via feed pipe 14B, will enter the combustion chamber 24 with a generally tangential component to promote cyclonic swirl in the combustion chamber 24; the air flow through these openings is marked with small arrows in Figure 4. Tangential velocities generally exceed 30 m/s and are typically about 50 m/s.

At the base of the Figure 4 combustor an outlet 33 is provided for ash. The cyclonic swirl within the combustion chamber 24 will cause the heavier ash and debris particles to be collected and exit via outlet 33, to be collected by an ash collection system (not shown).

The combustion of the biomass fuel and the secondary fuel in the combustion region 26 of the combustor 3 produces a hot pressurized combustion gas. The gas pressure and temperature in the combustion chamber 24 is typically 3-5 barg and 1100 to 1500°C respectively. In the Figure 4 embodiment, the hot pressurized combustion gas exits the combustion chamber 24 via an outlet in the form of a choke tube 34. Downstream of the outlet 34, before passing completely out of the combustor 3, the hot gas passes through a dilution chamber 3C. The circumference of the dilution chamber 3C is provided with a plurality of inlets for compressed air from compressed air feed 14A. Not only does the supply of compressed air to the dilution chamber 3C reduce the mean gas temperature from approximately 1200°C to approximately 800°C, but by appropriate shaping of the compressed air inlets in the dilution chamber 3C the air may be introduced into the dilution chamber in a manner so as to reduce the swirl of the exhaust gas. The reduced temperature combustion gas finally exits the combustor assembly 3 along pipe 5 to pass, via cyclone filter 9, to the inlet of the turbine section 7 of the gas turbine 6.

The near instantaneous (< 10 ms) fragmentation of the biomass particles in the combustor is highly desirable in that they may be made to burn with combustion kinetics similar to those of liquid fuel droplets. Furthermore, volatile and char burnout of the biomass fuel particles takes place within about 100 ms. Smaller wood particles burn more rapidly than larger wood particles. However, mechanically breaking wood particles up to make them smaller is demanding of both time and energy. By utilizing the rapid heating of the biomass particles in the secondary fuel flame and the radiant heat from biomass combustion, wood particles of a maximum dimension of 20 mm can be made to fragment near instantaneously (in less than 10 ms) to provide efficiently very small particles (notionally <<1 mm). A further important consequence of the small biomass particle size is that the ash particles that derive from it will also be small (typically < 1 micron) and can be made to pass between the blades of the turbine without deposition or impaction upon the blades, provided they can be prevented from agglomerating to form larger particles. This is achieved by rapidly cooling the particles in the dilution chamber before agglomeration can take place.

An alternative embodiment of combustor construction is illustrated in Figure 5. In many respects, this combustor is similar to that illustrated in Figure 4. The Figure 5 combustor also includes a metal flame can 30 surrounded by a wind box 31, with the chamber 32 formed therebetween being supplied with compressed air from the gas turbine's compressor section 8 (not shown). The walls of the metal flame can are provided with air injection apertures 70 to promote swirl in the combustor.

In the Figure 5 embodiment no ash collection exit (equivalent to outlet 33 in Figure 4) is provided, it can be provided separately as part of the cyclone filter 9 in Figure 1. The Figure 5 combustor does, however, include a downstream dilution chamber 3C, into which compressed air is introduced with a significant radial component to reduce the temperature of the hot pressurized combustion gas and to reduce swirl. The reduced temperature gas passes from the combustor 3 at the right hand end (as drawn in Figure 5) to exhaust pipe 5 and filter 9 (not shown).

The most significant difference between the embodiments of combustor illustrated in Figures 4 and 5 lies at the left hand end, in the arrangement of the secondary burner 28 and the biomass/air mixture inlet pipe 25. Although (as in the Figure 4 embodiment) in the Figure 5 embodiment the biomass/air mixture entry pipe 25 is generally tangential to the longitudinal axis of the combustor 3, in contrast to the Figure 4 embodiment the secondary burner is axially aligned with the longitudinal axis of the combustor so that its flame enters axially rather than tangentially as in the Figure 4 embodiment. The operation of both embodiments is, however, similar in that the wood/transport air mixture entering the combustion region 26 of both embodiments of combustor 3 is rapidly heated in the biomass flame and the flame of the secondary burner 28 to produce near instantaneous fragmentation of the biomass particles, and hence accelerated combustion rates for those particles.

The illustrated embodiments of combustor are capable of operating efficiently at low pressure ratios and with high excess air. They are thus able to provide acceptable turbine inlet temperatures, produce small size ash particles, allow for simple low cost pressure containment and have low environmental emission levels. For example, it is anticipated that the following levels are achievable: $CO < 200$ mg/MJ and $NOX < 90$ ppm at 15 % oxygen.

Although both illustrated embodiments of gas turbine have a turbine section including a single turbine stage and a compressor section including a single compressor stage, this is not essential. It is thought that more complicated, multi-stage gas turbine constructions may be possible while retaining the robust and flexible characteristics of the above described embodiments of gas turbine and avoiding the undesirable characteristics typical of aero-derived, or high performance land-based, gas turbines.

Although it is preferred to operate the turbine in a recuperated cycle, in which case the turbine will work at maximum efficiency at a turbine section gas inlet pressure in the range of 3-7 times atmospheric pressure, the turbine is not restricted to being used in a recuperated cycle.

CLAIMS

1. A biomass fuel combustor for use in the pressurised combustion of biomass fuel particles to produce a pressurised exhaust gas, the combustor comprising:

a cyclonic combustion chamber having an upstream combustion region and a downstream combustion region, the upstream region having first and second fuel inlets, the second inlet being for the entry into the upstream region of biomass fuel particles for combustion and the first inlet being for the entry into the upstream region of gas and/or liquid secondary fuel for combustion in the upstream region at least during start-up of the combustor to supply heat to the upstream region to promote fragmentation in the upstream region of the biomass fuel particles entering the upstream region via the second inlet,

wherein the second inlet is positioned such that, in use, biomass fuel particles entering the upstream region via the second inlet will do so with a tangential component to promote cyclonic motion of gases and biomass particles in the upstream region and wherein the downstream region of the combustion chamber is provided with a plurality of air injection openings arranged such that air will, in use, enter the downstream region of the combustion chamber with a tangential component to promote cyclonic motion of gases and fragmented biomass particles in the downstream region of the combustion chamber.

2. A combustor as claimed in claim 1, wherein the combustor is so constructed and arranged that biomass fuel particle fragmentation will take place within 10 ms of said particles entering the upstream region of the combustor via the second inlet.

3. A combustor as claimed in claim 1 or claim 2, wherein the combustion chamber is so constructed and arranged that both volatile and char burnout of the biomass fuel particles will be complete within 100 ms of the particles entering the upstream region of the combustor via the second inlet.

4. A combustor as claimed in any one of the preceding claims, wherein

the combustion chamber is substantially defined by a flame can.

5. A combustor as claimed in claim 4, wherein the flame can is predominantly made of metal.

6. A combustor as claimed in claim 4 or claim 5, wherein the flame can, at least in the region of the first and second fuel inlets, is provided internally with a refractory material layer of less than 2 mm or is unlined with refractory material.

7. A combustor as claimed in any one of claims 4 to 6, wherein the flame can is substantially enclosed by a wind box, to form a chamber between the exterior of the flame can and the interior of the wind box, whereby a supply of pressurized cooling air to said chamber will act to cool the flame can.

8. A combustor as claimed in claim 7, wherein the air injection openings are provided in the wall of the combustion chamber, opening into the wind box, so that the pressurized cooling air supplied to the chamber between the combustion chamber and the wind box will, in use, enter the combustion chamber via said openings.

9. A combustor as claimed in any one of the preceding claims, wherein the first inlet forms part of a burner for the secondary fuel.

10. A combustor as claimed in claim 9, further comprising a secondary fuel feed controller for regulating the feed rate of secondary fuel to the burner.

11. A combustor as claimed in claim 10, wherein the burner is for liquid or gaseous fossil or bio-fuel.

12. A combustor as claimed in any one of the preceding claims, wherein the first inlet is positioned relative to the cyclonic combustion chamber such that, in use, the flame produced by combustion of the secondary fuel will be generally tangential to the longitudinal axis of the chamber.

13. A combustor as claimed in claim 12, wherein the first and second inlets are generally diametrically opposite one another.

14. A combustor as claimed in any one of claims 1 to 11, wherein the first inlet is generally coaxial with the longitudinal axis of the combustion chamber.

15. A combustor as claimed in any one of the preceding claims, wherein

the combustor is provided with a first outlet for the efflux from the combustion chamber of the hot pressurized combustion gas resulting from the biomass and/or secondary fuel combustion.

16. A combustor as claimed in any one of the preceding claims, further comprising a dilution chamber downstream of said first outlet, said dilution chamber being provided with one or more third inlets for the supply to the dilution chamber of air to dilute the hot pressurized combustion gas.

17. A combustor as claimed in claim 16, wherein a plurality of individual third inlets are provided and are arranged to introduce air into the dilution chamber in a manner so as to reduce the cyclonic motion and temperature of the hot pressurized combustion gas passing therethrough.

18. A combustor as claimed in any one of claims 16 and 17, wherein the combustion chamber comprises a second outlet to said dilution chamber, for the removal of ash.

19. A direct cycle biomass-fired gas turbine system, comprising:

a biomass fuel combustor as claimed in any one of claims 1 to 18; and

a gas turbine including a turbine section, arranged to receive and be driven by the hot pressurized combustion gas resulting from said biomass combustion, and a compressor section.

20. A system as claimed in claim 19, wherein the gas turbine is so constructed and arranged that, when operated in a recuperated cycle at a turbine section gas inlet pressure in the range of 3-7 times atmospheric pressure, the gas turbine will work at maximum efficiency.

21. A system as claimed in claim 19 or claim 20, wherein the turbine section comprises a stator and a rotor that are tolerant to ash present in the hot pressurized combustion gas, with passages between the blades of the stator and the rotor being configured so as to minimize deposition and impaction of ash particles of the size present in the hot pressurized combustion gas.

22. A system as claimed in any one of claims 19 to 21, wherein the turbine section is of the radial flow type.

23. A system as claimed in any one of claims 19 to 22, wherein the gas turbine is derived from an adapted automotive, marine or similar turbocharger.

24. A system as claimed in any one of claims 19 to 23, wherein the gas turbine is so constructed and arranged as to work at high efficiency at a turbine section inlet gas temperature of less than 900°C.

25. A system as claimed in any one of claims 19 to 24, wherein the turbine section is arranged to receive the hot pressurized combustion gas directly from the combustor.

26. A system as claimed in any one of claims 19 to 25, wherein the gas turbine includes a drive shaft connecting its turbine and compressor sections.

27. A system as claimed in claim 26, wherein the gas inlet to the turbine section is not coaxial with the axis of said shaft.

28. A system as claimed in claim 26 or 27, wherein the turbine includes only a single said drive shaft.

29. A system as claimed in any one of claims 26 to 28, wherein the gas inlet to the turbine section is generally tangential to said axis.

30. A system as claimed in any one of claims 19 to 29, wherein said turbine section includes a single turbine stage.

31. A system as claimed in claim 30, wherein said turbine section includes a radial inflow turbine wheel.

32. A system as claimed in any one of claims 19 to31, wherein said compressor section includes a single compressor stage.

33. A system as claimed in claim 32, wherein the compressor section includes a centrifugal compressor wheel.

34. A system as claimed in any one of claims 19 to 29, wherein at least one of said turbine and compressor sections is multi-staged.

35. A method of pressurized combustion of biomass fuel to produce power, the method comprising:

providing a system as claimed in any one of claims 19 to 34;

combusting biomass fuel under pressure in the combustor to produce hot pressurized combustion gas;

expanding said hot pressurized combustion gas through the turbine section of the gas turbine to produce a power output.

36. A method as claimed in claim 35, further comprising supplying a secondary gas and/or liquid fuel to the combustion chamber of the combustor.

37. A method of pressurised combustion of biomass fuel in a biomass fuel combustor to produce a pressurised exhaust gas, the method comprising:

providing a biomass fuel combustor having a cyclonic combustion chamber;

supplying gas and/or liquid secondary fuel to an upstream combustion region of said combustion chamber;

combusting said secondary fuel in said upstream region of said combustion chamber;

supplying biomass fuel particles to said upstream region of said combustion chamber with a tangential component to promote cyclonic motion of gases and biomass particles in said upstream region;

at least during start-up of the combustor fragmenting said supplied biomass fuel particles in said upstream region using the heat generated by the combustion of said secondary fuel in said upstream region;

passing the fragmented biomass particles from said upstream region to a downstream region of said combustion chamber;

adding air into said downstream region with a tangential component to promote cyclonic motion of gases and fragmented biomass particles in said downstream region; and

completing combustion of said fragmented biomass particles in said downstream region.

38. A method as claimed in claim 37, wherein said combustion of said biomass particles produces hot pressurised combustion gas, which gas is expanded through the turbine section of a gas turbine to produce a power output.

39. A method as claimed in any one of claims 36 to 38, wherein the secondary fuel is used for start-up of the process, for supporting sustained ignition of

the biomass fuel in the combustion chamber of the combustor and for process control purposes.

40. A method as claimed in any one of claimed 36 to 39, wherein on start-up of the process only secondary fuel is supplied to the combustion chamber, with feeding of the biomass fuel being commenced only after combustion in the chamber has been established with the secondary fuel.

41. A method as claimed in any one of claims 36 to 40, wherein at start-up, as the rate of feed of biomass fuel to the combustor is increased, the rate of supply of secondary fuel is reduced.

42. A method as claimed in any one of claims 36 to 41, wherein the rate of supply of secondary fuel is varied in response to variations in turbine shaft speed and power output demand.

43. A method as claimed in any one of claims 35, 36 and 38 to 42, wherein the hot pressurized combustion gas is supplied to the turbine section at a temperature of approximately 700°C-900°C.

44. A method as claimed in any one of claims 35, 36 and 38 to 43, wherein the hot pressurized combustion gas is supplied to the turbine section at a pressure of approximately 3-7 times atmospheric pressure.

45. A method as claimed in any one of claims 35, 36 and 38 to 44, wherein the hot pressurized combustion gas is supplied to the turbine section at a temperature below the ash fusion temperature of the biomass fuel.

46. A method as claimed in claim 45, wherein the ash fusion temperature is greater than 800°C.

47. A method as claimed in any one of claims 35 to 46, wherein the pressure in the combustor is in the range of approximately 3-7 times atmospheric pressure.

48. A method as claimed in claim 35 or claim 38, wherein the turbine section is tolerant to deposition, erosion and corrosion from the exhaust gas.

49. A method as claimed in any one of claims 35 to 48, wherein the biomass fuel is supplied to the combustion chamber of the combustor from a fuel

supply at atmospheric pressure through a valve operated to maintain the higher pressure in the combustion chamber.

50. A method as claimed in any one of claims 35 to 49, wherein air is supplied to the combustion chamber at a rate approximately twice that necessary for stoichiometric combustion.

51. A method as claimed in any one of claims 35 to 40, wherein the heat release rate of the combusted biomass fuel is greater than 5 MW/m^3 of biomass fuel.

52. A method as claimed in claim 51, wherein the heat release rate is such that both volatile and char burnout of the biomass fuel is complete within 100 ms.

53. A method as claimed in any one of claims 35 to 52, wherein the mean temperature at the outlet from the combustion chamber of the combustor is typically between 1100°C and 1500°C

54. A method as claimed in any one of claims 35 to 53 when dependant from claim 17, wherein the temperature of the hot pressurized combustion gas exiting the dilution chamber is typically between 700°C and 900°C.

55. A method as claimed in any one of claims 35 to 54, wherein ash is removed by passing the hot pressurized combustion gas through a filter between the combustor and the gas turbine.

56. A method as claimed in any one of claims 35 to 55, wherein the system can run and produce full power when solely fuelled by the secondary fuel.

57. A method as claimed in any one claims 35 to 56, wherein the gas turbine is used to generate electrical power.

58. A method as claimed in any one of claims 35 to 57, wherein the compressor section of the gas turbine is used to compress air, at least some of the compressed air so produced being supplied to the combustor with the biomass fuel .

59. A method as claimed in claim 58, wherein said compressed air is also used to cool the combustor.

60. A method as claimed in any one of claims 35 to 59, further comprising taking the expanded waste gas from the turbine section's exit and using said waste gas to heat compressed air output by the gas turbine's compressor section

and/or in a heat recovery heat exchanger as part of a CHP system and/or for direct use as part of a drying system.

61. A method as claimed in any one of claims 35 to 60, wherein the ash particles produced by the biomass particle combustion are of a sufficiently small size that they will pass through the turbine section without significant deposition, erosion or corrosion damage to the blades of the turbine section.

62. A method as claimed in claim 61, wherein the ash particles produced by the biomass particle combustion have a mean size of < 1 micron.

63. A process as claimed in any one of claims 35 to 62, wherein the biomass particles supplied to the combustor are wood chips having a particle size of maximum dimension up to 20 mm, passing through a mesh of 10 mm maximum.

64. A process as claimed in any one of claims 35 to 63, wherein the biomass fuel particles fragment within 10 ms of the particles entering the combustion chamber of the combustor.

65. A process as claimed in claim 64, wherein the particles fragment to a mean size of << 1 mm.

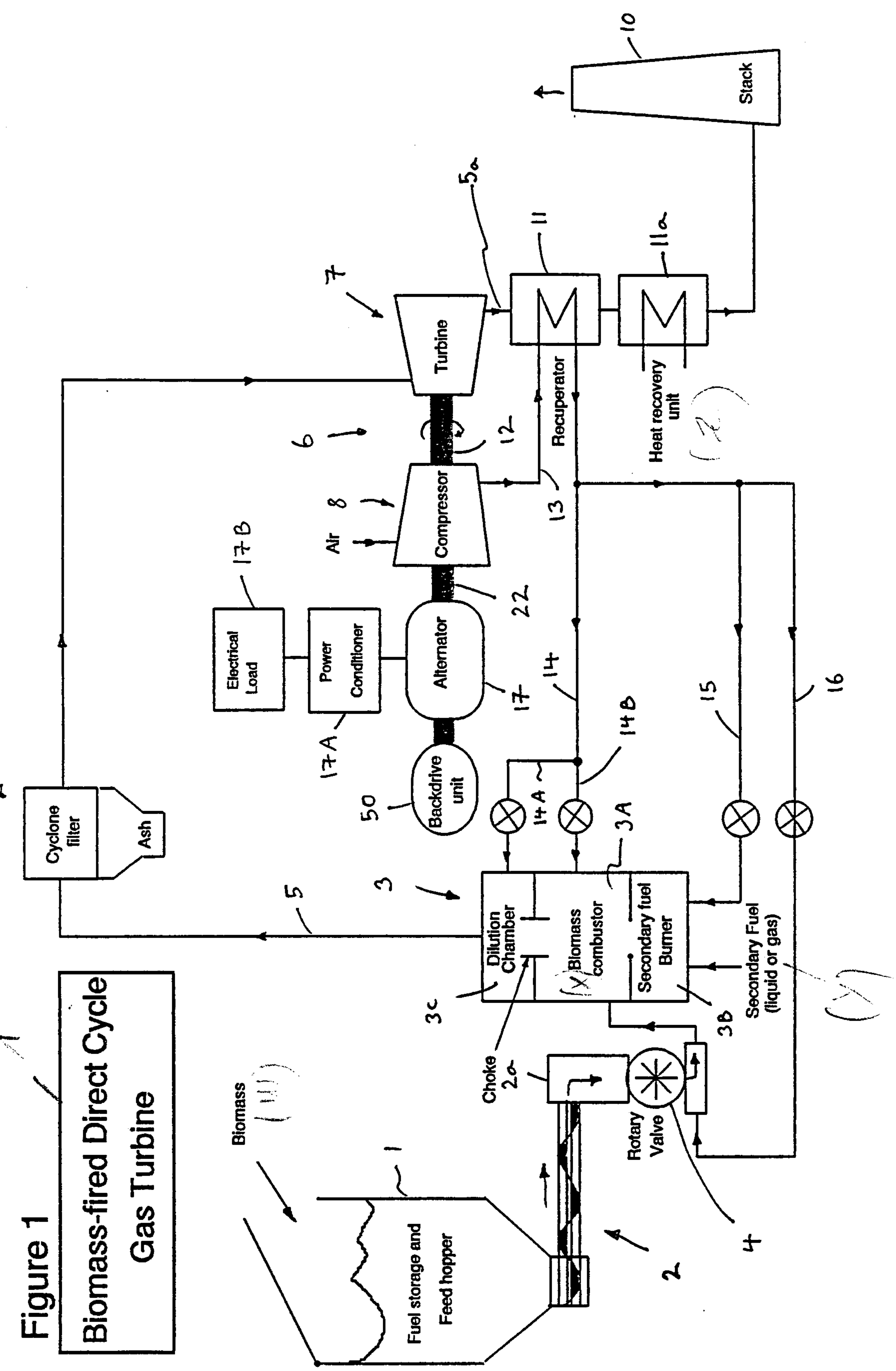


Figure 1
Biomass-fired Direct Cycle Gas Turbine

FIGURE 2

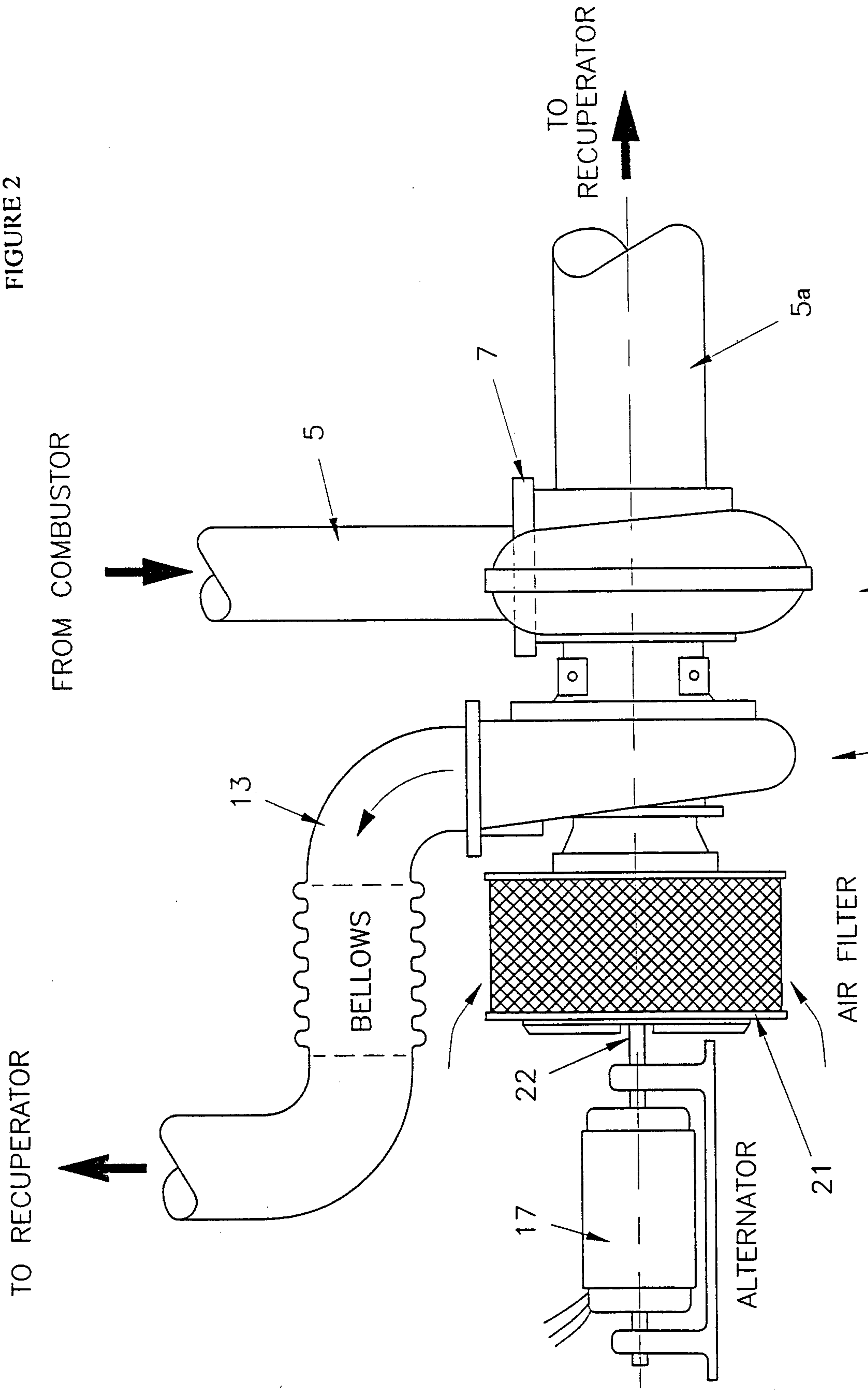

TO RECUPERATOR
FROM COMBUSTOR
5
7
5a
6
8
13
BELLOWS
AIR FILTER
22
21
17
ALTERNATOR
TO RECUPERATOR

FIGURE 3

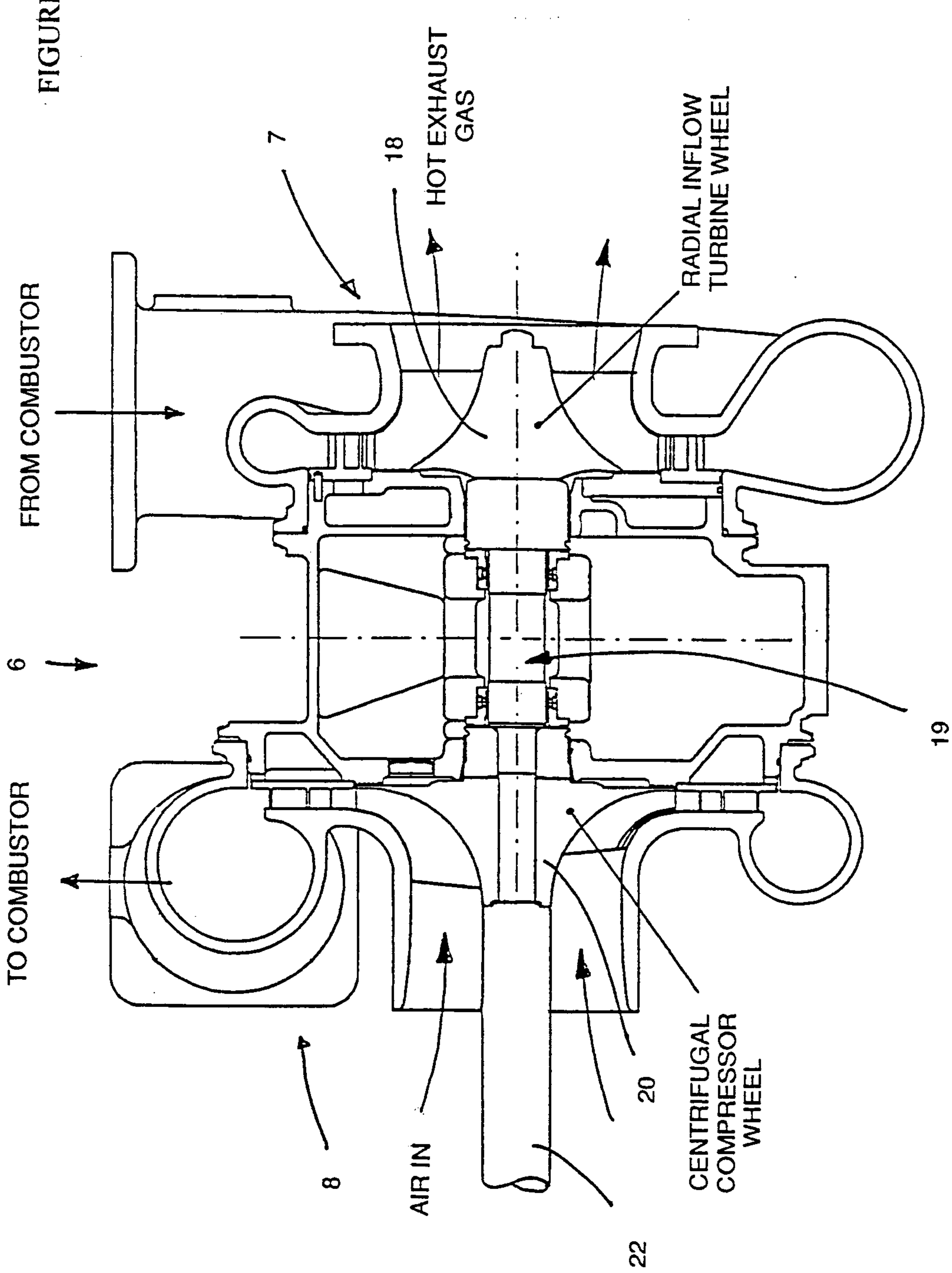

FROM COMBUSTOR
7
18
HOT EXHAUST GAS
RADIAL INFLOW TURBINE WHEEL
6
19
TO COMBUSTOR
8
AIR IN
22
20
CENTRIFUGAL COMPRESSOR WHEEL

FIGURE 4

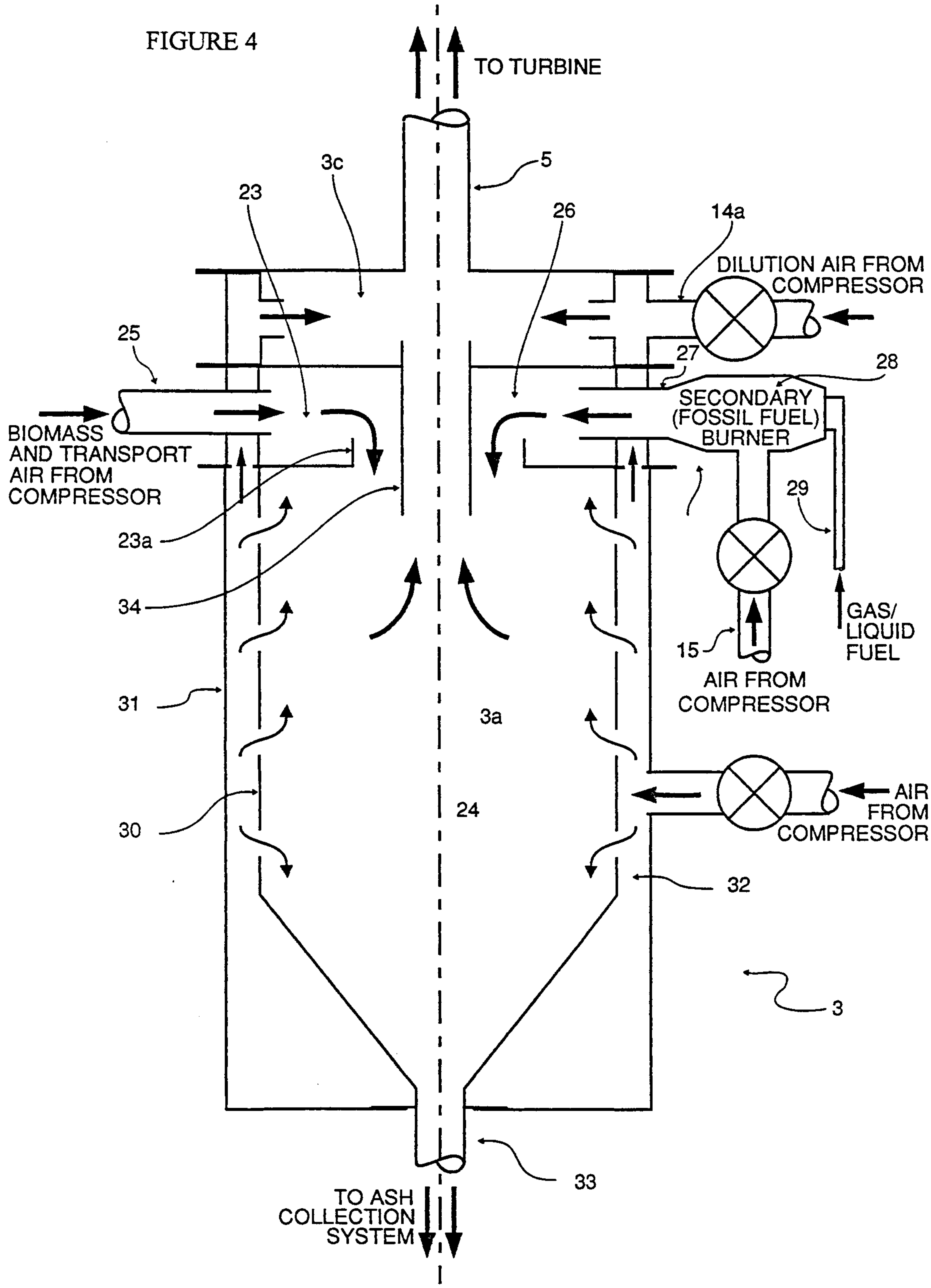
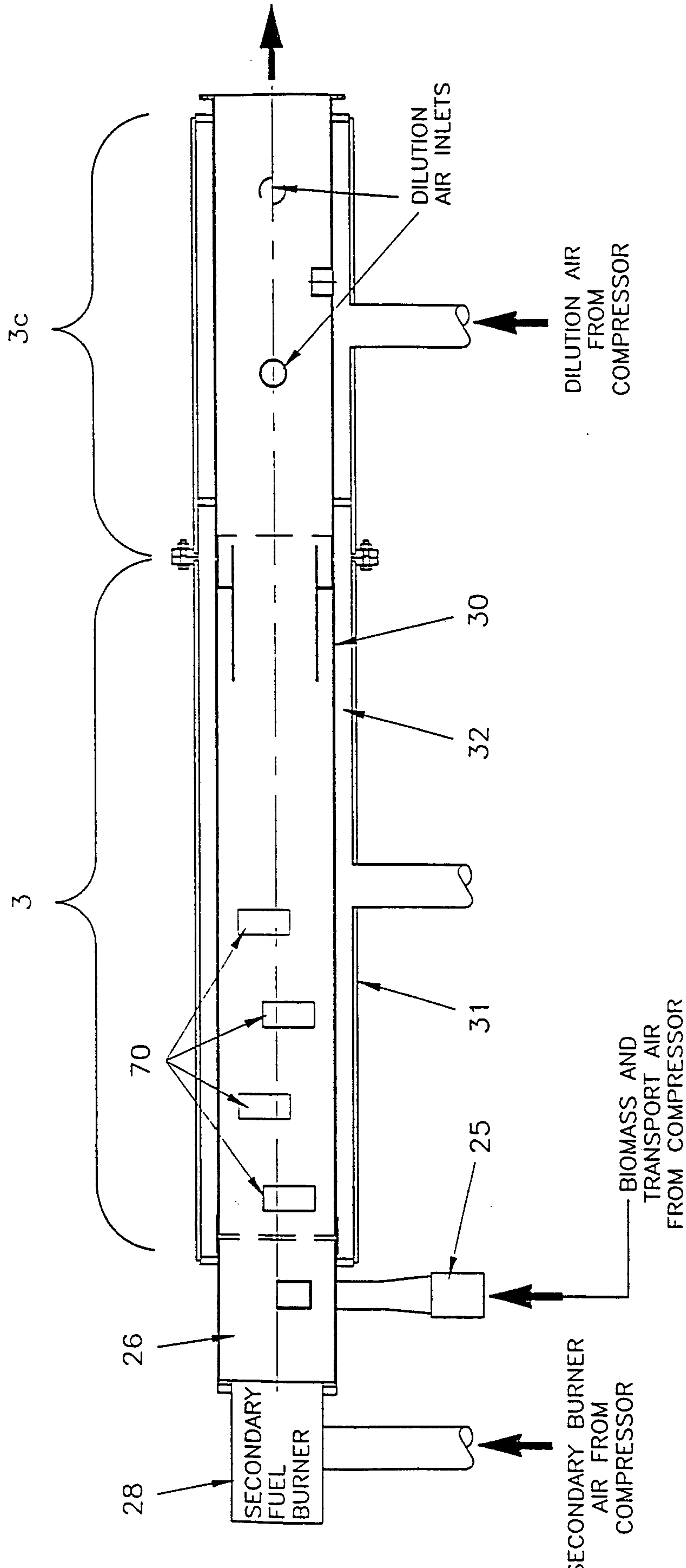
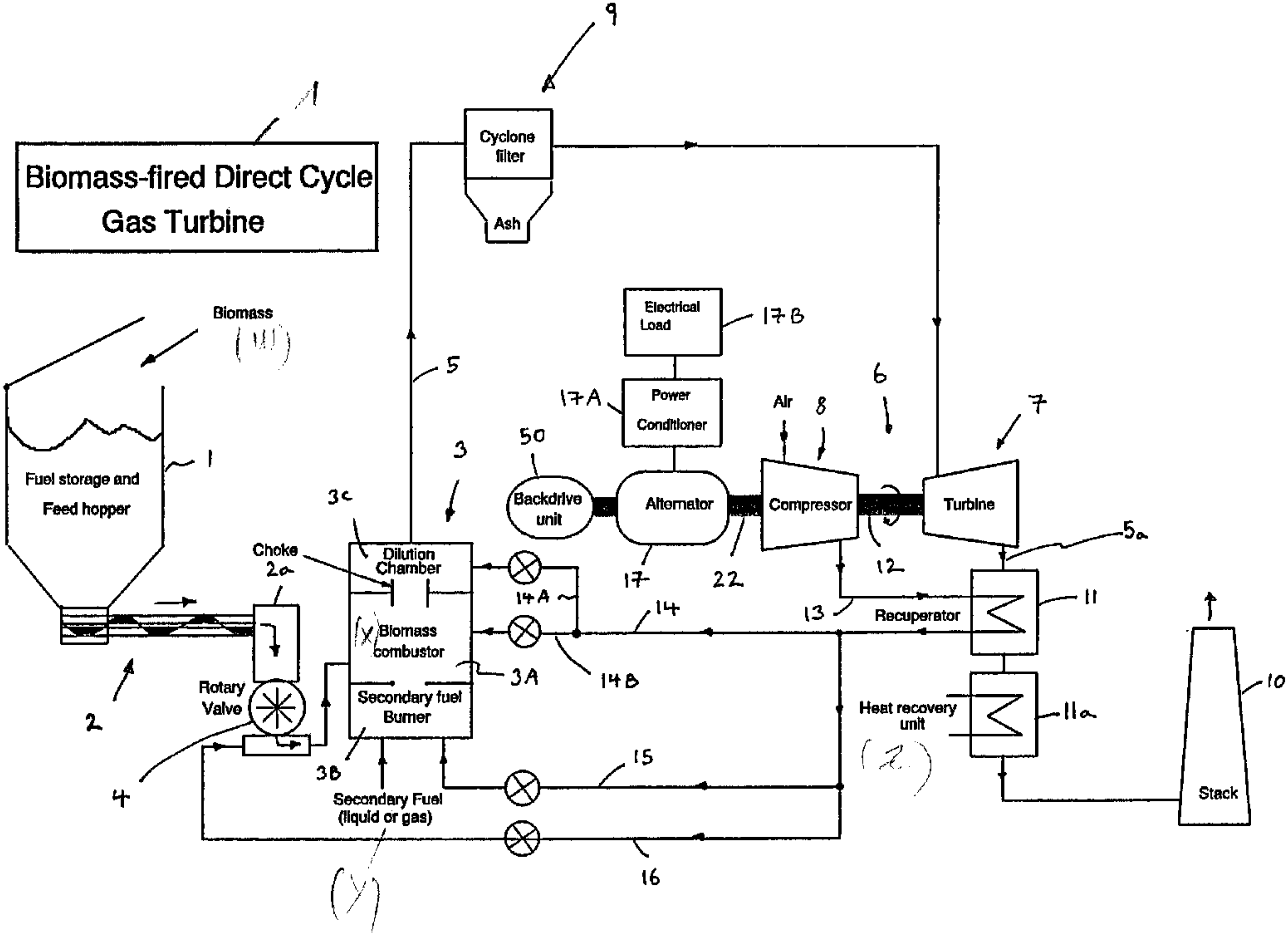

TO TURBINE
3c
5
23
26
14a
DILUTION AIR FROM COMPRESSOR
25
27
28
SECONDARY (FOSSIL FUEL) BURNER
BIOMASS AND TRANSPORT AIR FROM COMPRESSOR
29
23a
34
GAS/ LIQUID FUEL
15
AIR FROM COMPRESSOR
31
3a
AIR FROM COMPRESSOR
30
24
32
3
33
TO ASH COLLECTION SYSTEM

FIGURE 5


DILUTION AIR INLETS
DILUTION AIR FROM COMPRESSOR
3c
3
30
32
31
70
25
BIOMASS AND TRANSPORT AIR FROM COMPRESSOR
26
SECONDARY FUEL BURNER
28
SECONDARY BURNER AIR FROM COMPRESSOR



Biomass-fired Direct Cycle Gas Turbine
1
Biomass
(W)
Fuel storage and Feed hopper
1
2
Choke
2a
Rotary Valve
4
Cyclone filter
Ash
9
5
3
3c
Dilution Chamber
(X) Biomass combustor
3A
Secondary fuel Burner
3B
Secondary Fuel (liquid or gas)
(Y)
Electrical Load
17B
Power Conditioner
17A
50
Backdrive unit
Alternator
17
22
Air
8
Compressor
6
12
7
Turbine
5a
13
Recuperator
11
Heat recovery unit
11a
(Z)
14A
14
14B
15
16
Stack
10