Office de la Propriété
Intellectuelle
du Canada

Un organisme
d'Industrie Canada

Canadian
Intellectual Property
Office

An agency of
Industry Canada

CA 2400599 A1 2001/09/07

(21) 2 400 599

(12) DEMANDE DE BREVET CANADIEN
CANADIAN PATENT APPLICATION

(13) A1

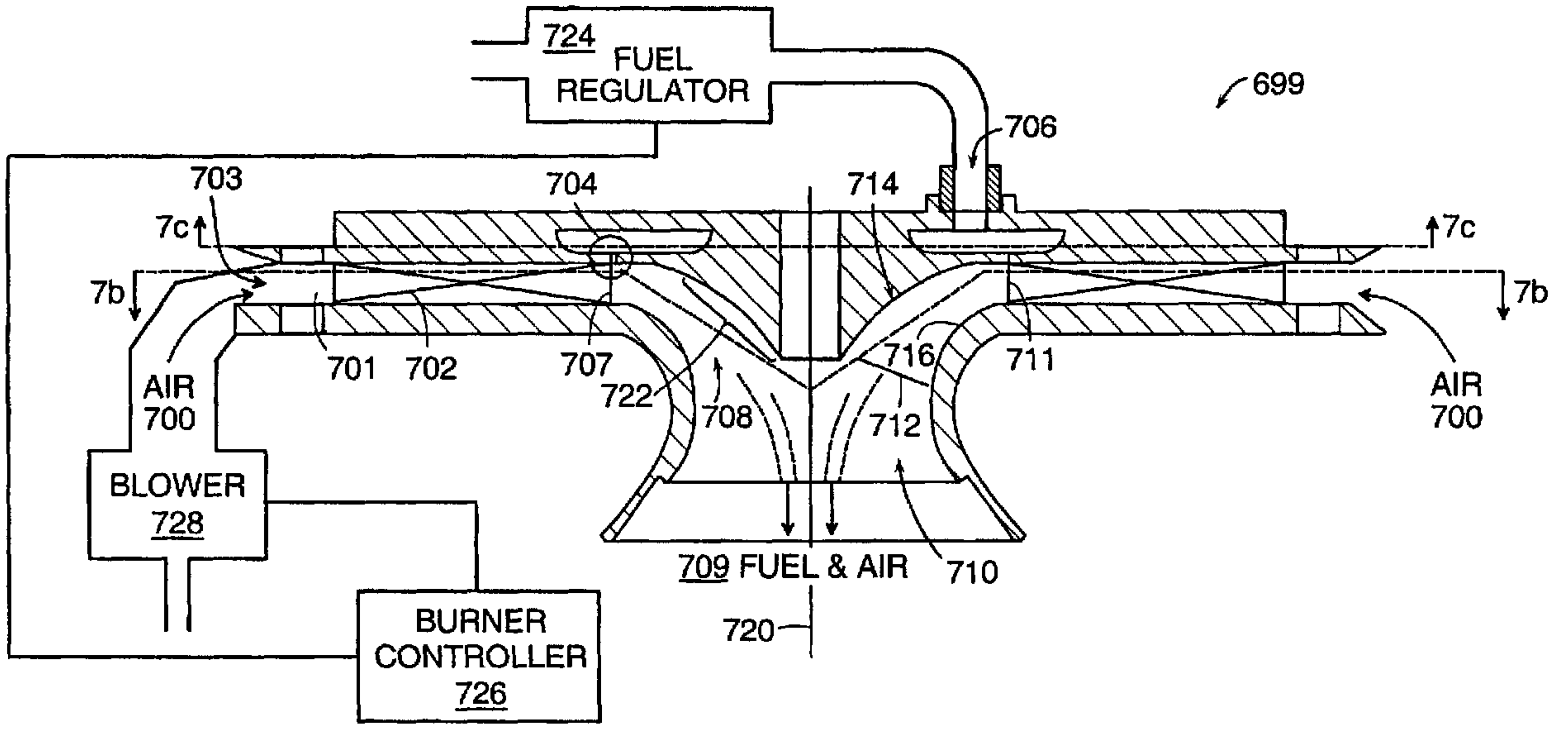
(86) **Date de dépôt PCT/PCT Filing Date**: 2001/03/01
(87) **Date publication PCT/PCT Publication Date**: 2001/09/07
(85) **Entrée phase nationale/National Entry**: 2002/08/16
(86) **N° demande PCT/PCT Application No.**: US 2001/040201
(87) **N° publication PCT/PCT Publication No.**: 2001/065102
(30) **Priorité/Priority**: 2000/03/02 (09/517,686) US

(51) **Cl.Int.^{7}/Int.Cl.^{7}** F02G 5/00
(71) **Demandeur/Applicant:**
NEW POWER CONCEPTS LLC, US

(72) **Inventeurs/Inventors:**
KAMEN, DEAN L., US;
NORRIS, MICHAEL, US;
LANGENFELD, CHRISTOPHER C., US

(74) **Agent**: GOWLING LAFLEUR HENDERSON LLP

(54) Titre : SYSTEME ET PROCEDE DE CONTROLE DE LA DIFFUSION DE COMBUSTIBLE ET D'AIR DANS UN BRULEUR DE MOTEUR A CYCLE THERMIQUE
(54) Title: SYSTEM AND METHOD FOR CONTROL OF FUEL AND AIR DELIVERY IN A BURNER OF A THERMAL-CYCLE ENGINE



(57) **Abrégé/Abstract:**
A method of combusting fuel and air in a burner of an external combustion engine having a heater head. The fuel and air are combined to form a fuel-air mixture which is characterized by a fuel-air ratio. An exhaust gas product is produced when the fuel-air mixture is combusted in the burner of the external combustion engine. A flame is formed by igniting the fuel-air mixture at a first fuel-air ratio produced by a first air flow rate and a fuel flow rate. The air flow rate is then increased to produce a second fuel-air ratio. The fuel flow rate is also controlled based upon a temperature of the heater head of the external combustion engine. The flame is maintained at the second fuel-air ratin adjusting the air flow rate based on the fuel flow rate. The external combustion engine may be, for example, a Stirling cycle engine.

(12) INTERNATIONAL APPLICATION PUBLISHED UNDER THE PATENT COOPERATION TREATY (PCT)

(19) **World Intellectual Property Organization**
International Bureau

(43) **International Publication Date**
7 September 2001 (07.09.2001)

PCT

(10) **International Publication Number**
WO 01/65102 A1

(51) **International Patent Classification⁷:** **F02G 5/00**

(21) **International Application Number:** PCT/US01/40201

(22) **International Filing Date:** 1 March 2001 (01.03.2001)

(25) **Filing Language:** English

(26) **Publication Language:** English

(30) **Priority Data:**
09/517,686 2 March 2000 (02.03.2000) US

(71) **Applicant: NEW POWER CONCEPTS LLC** [US/US]; 340 Commercial Street, Manchester, NH 03101 (US).

(72) **Inventors: KAMEN, Dean, L.**; 15 Westwind Road, Bedford, NH 03110 (US). **NORRIS, Michael**; 95 Mirror Street, Manchester, NH 03104 (US). **LANGENFELD, Christopher, C.**; 4 Dunloggin Road, Nashua, NH 03063 (US).

(74) **Agents: TIBBETTS, Jean, M.** et al.; Bromberg & Sunstein LLP, 125 Summer Street, Boston, MA 02110 (US).

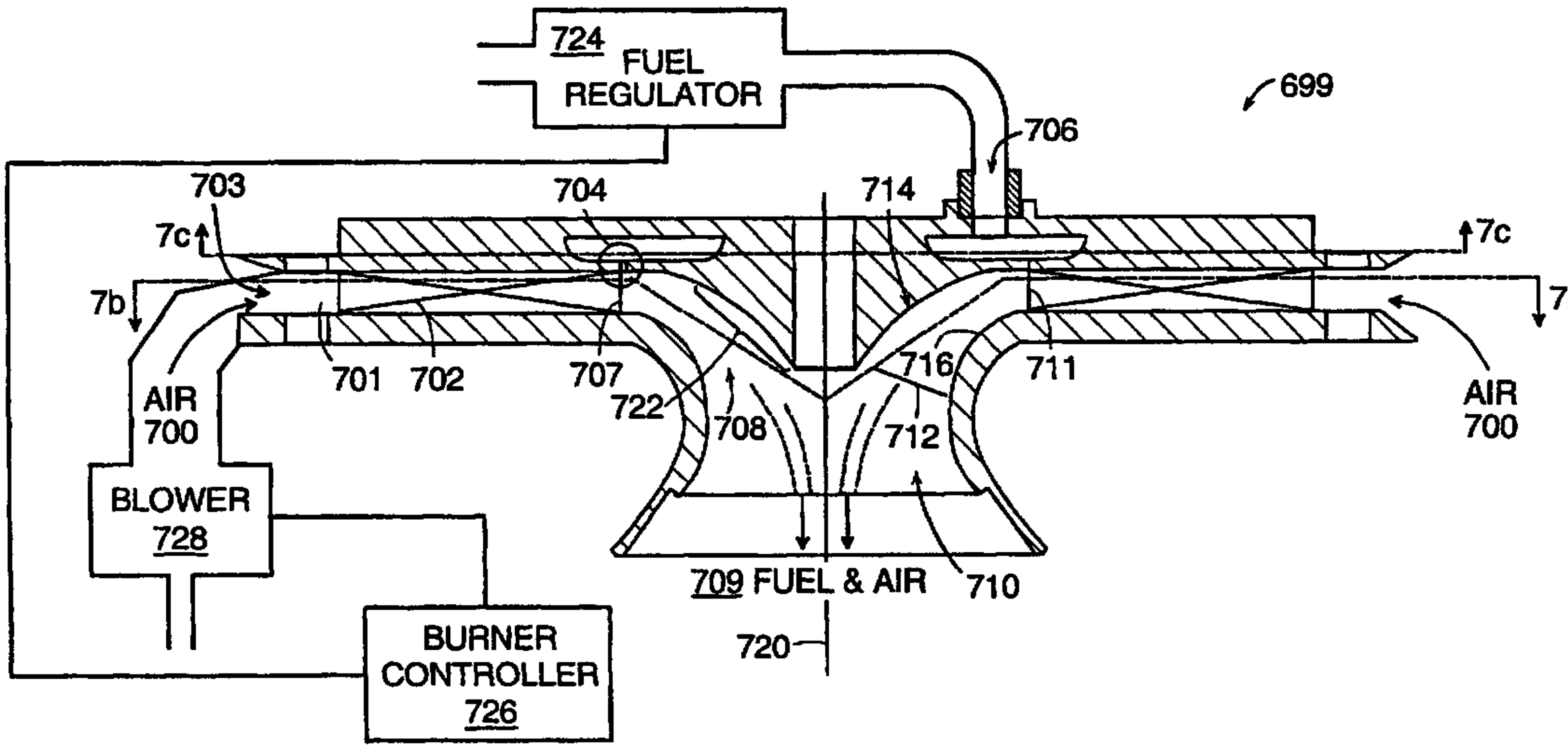
(81) **Designated States** *(national)*: AE, AG, AL, AM, AT, AU, AZ, BA, BB, BG, BR, BY, BZ, CA, CH, CN, CO, CR, CU, CZ, DE, DK, DM, DZ, EE, ES, FI, GB, GD, GE, GH, GM, HR, HU, ID, IL, IN, IS, JP, KE, KG, KP, KR, KZ, LC, LK, LR, LS, LT, LU, LV, MA, MD, MG, MK, MN, MW, MX, MZ, NO, NZ, PL, PT, RO, RU, SD, SE, SG, SI, SK, SL, TJ, TM, TR, TT, TZ, UA, UG, UZ, VN, YU, ZA, ZW.

(84) **Designated States** *(regional)*: ARIPO patent (GH, GM, KE, LS, MW, MZ, SD, SL, SZ, TZ, UG, ZW), Eurasian patent (AM, AZ, BY, KG, KZ, MD, RU, TJ, TM), European patent (AT, BE, CH, CY, DE, DK, ES, FI, FR, GB, GR, IE, IT, LU, MC, NL, PT, SE, TR), OAPI patent (BF, BJ, CF, CG, CI, CM, GA, GN, GW, ML, MR, NE, SN, TD, TG).

Published:
— *with international search report*
— *before the expiration of the time limit for amending the claims and to be republished in the event of receipt of amendments*

For two-letter codes and other abbreviations, refer to the "Guidance Notes on Codes and Abbreviations" appearing at the beginning of each regular issue of the PCT Gazette.

(54) **Title:** SYSTEM AND METHOD FOR CONTROL OF FUEL AND AIR DELIVERY IN A BURNER OF A THERMAL-CYCLE ENGINE



(57) **Abstract:** A method of combusting fuel and air in a burner of an external combustion engine having a heater head. The fuel and air are combined to form a fuel-air mixture which is characterized by a fuel-air ratio. An exhaust gas product is produced when the fuel-air mixture is combusted in the burner of the external combustion engine. A flame is formed by igniting the fuel-air mixture at a first fuel-air ratio produced by a first air flow rate and a fuel flow rate. The air flow rate is then increased to produce a second fuel-air ratio. The fuel flow rate is also controlled based upon a temperature of the heater head of the external combustion engine. The flame is maintained at the second fuel-air ratin adjusting the air flow rate based on the fuel flow rate. The external combustion engine may be, for example, a Stirling cycle engine.

WO 01/65102 A1

System and Method for Control of Fuel and Air Delivery in a Burner of a Thermal-Cycle Engine

Technical Field

The present invention pertains to improvements to a Stirling cycle heat engine or refrigerator and more particularly to improvements relating to mechanical and thermal components of a Stirling cycle heat engine or refrigerator which contribute to increased engine operating efficiency and lifetime, and to reduced size, complexity and cost.

Background of the Invention

Stirling cycle machines, including engines and refrigerators, have a long technological heritage, described in detail in Walker, *Stirling Engines*, Oxford University Press (1980), incorporated herein by reference. The principle underlying the Stirling cycle engine is the mechanical realization of the Stirling thermodynamic cycle: isovolumetric heating of a gas within a cylinder, isothermal expansion of the gas (during which work is performed by driving a piston), isovolumetric cooling, and isothermal compression. The Stirling cycle refrigerator is also the mechanical realization of a thermodynamic cycle which approximates the ideal Stirling thermodynamic cycle. In an ideal Stirling thermodynamic cycle, the working fluid undergoes successive cycles of isovolumetric heating, isothermal expansion, isovolumetric cooling and isothermal compression. Practical realizations of the cycle, wherein the stages are neither isovolumetric nor isothermal, are within the scope of the present invention and may be referred to within the present description in the language of the ideal case without limitation of the scope of the invention as claimed.

Various aspects of the present invention apply to both Stirling cycle engines and Stirling cycle refrigerators, which are referred to collectively as Stirling cycle machines in the present description and in any appended claims.

The principle of operation of a Stirling cycle engine is readily described with reference to FIGS. 1a-1f, wherein identical numerals are used to identify the same or similar parts. Many mechanical layouts of Stirling cycle engines are known in the art, and the particular Stirling engine designated generally by numeral **10** is shown merely for illustrative purposes. In FIGS. la to ld, a piston **12** (otherwise referred to herein as a "compression piston") and a second piston (also known as an "expansion piston") **14** move in phased reciprocating motion within cylinder **16**. Compression piston **12** and expansion piston **14** may also move within separate, interconnected, cylinders. Piston seals **18** prevents the flow of a working fluid contained within cylinder **16** between piston **12** and piston **14** from escaping around either piston **12**. The working fluid is chosen for its thermodynamic properties, as discussed in the description below, and is typically helium at a pressure of several atmospheres. The volume of fluid governed by the position of expansion piston **14** is referred to as expansion space **22**. The volume of fluid governed by the position of compression piston **12** is referred to as compression space **24**. In order for fluid to flow between expansion space **22** and compression space **24**, whether in the configuration shown or in another configuration of Stirling engine **10**, the fluid passes through regenerator **26**. Regenerator **26** is a matrix of material having a large ratio of surface area to volume which serves to absorb heat from the working fluid when the fluid enters hot from expansion space **22** and to heat the fluid when it passes from compression space **24** returning to expansion space **22**.

During the first phase of the engine cycle, the starting condition of which is depicted in FIG. la, piston **12** compresses the fluid in compression space **24**. The compression occurs at a constant temperature because heat is extracted from the fluid to the ambient environment. In practice, a cooler **68** (shown in FIG. 2) is provided, as will be discussed in the description below.

The condition of engine **10** after compression is depicted in FIG. 1b. During the second phase of the cycle, expansion piston **14** moves in synchrony with compression piston **12** to maintain a constant volume of fluid. As the fluid is transferred to expansion space **22**, it flows through regenerator **26** and acquires heat from regenerator **26** such that the pressure of the fluid increases. At the end of the transfer phase, the fluid is at a higher pressure and is contained within expansion space **22**, as depicted in FIG. lc.

During the third (expansion) phase of the engine cycle, the volume of expansion space **22** increases as heat is drawn in from outside engine **10**, thereby converting heat to work. In practice, heat is provided to the fluid in expansion space **22** by means of a heater **64** (shown in FIG. 2) which is discussed in greater detail in the description below. At the end of the expansion phase, the hot fluid fills the full expansion space **22** as depicted in FIG. 1d. During the fourth phase of the engine cycle, the fluid is transferred from expansion space **22** to compression space **24**, heating regenerator **26** as the fluid passes through it. At the end of the second transfer phase, the fluid is in compression space **24**, as depicted in FIG. 1a, and is ready for a repetition of the compression phase. The Stirling cycle is depicted in a P-V (pressure-volume) diagram as shown in FIG. le and in a T-S (temperature -entropy) diagram as shown in FIG. lf. The Stirling cycle is a closed cycle in that the working fluid is typically not replaced during the course of the cycle.

The principle of operation of a Stirling cycle refrigerator can also be described with reference to FIGS. 1a-1e, wherein identical numerals are used to identify the same or similar parts. The differences between the engine described above and a Stirling machine employed as a refrigerator are that compression volume **22** is typically in thermal communication with ambient temperature and expansion volume **24** is connected to an external cooling load (not shown). Refrigerator operation requires net work input.

Stirling cycle engines have not generally been used in practical applications, and Stirling cycle refrigerators have been limited to the specialty field of cryogenics, due to several daunting engineering challenges to their development. These involve such practical considerations as efficiency, vibration, lifetime, and cost. The instant invention addresses these considerations.

Summary of the Invention

A method of combusting a fuel and air in a burner of an external combustion engine, the fuel and air combined to form a fuel-air mixture having a variable fuel-air ratio, the fuel-air mixture when combusted producing an exhaust gas product includes igniting the fuel-air mixture to form a flame at a first fuel-air ratio produced by a first air flow rate and a fuel flow rate, increasing the air flow rate to produce a second fuel-air ratio, controlling the fuel flow rate based at least on a temperature of the heater head, and maintaining the flame at the second fuel-air ratio by adjusting the air flow rate based at least on a temperature of the air and an oxygen concentration in the exhaust gas product. Igniting the fuel and air where the fuel having an auto-ignition temperature and a flame speed includes propelling the air at a speed above the flame speed into an inlet of a throat, the throat also having an outlet and a constant cross sectional area from inlet to outlet and mixing fuel into the air forming the fuel-air mixture, the fuel-air mixture exiting the outlet, such that a flame is created in the air fuel mixture outside the outlet of the throat.

In accordance with another embodiment of the invention, the second fuel-air ratio is maintained by adjusting the air flow rate based on an oxygen concentration in the exhaust gas. In a further embodiment, the second fuel-air ratio may be maintained by adjusting the air flow rate based at least on a temperature of the air and the fuel flow rate. In an alternative embodiment, the second fuel-air ratio is maintained by adjusting the air flow rate based at least on a temperature of the air and an oxygen concentration in the exhaust gas.

A system for operating a combustion chamber of an external combustion engine having a heater head, the combustion chamber characterized by a combustion axis and for delivering heat to the heater head of the engine by combusting a fuel in air to produce heat and an exhaust gas product includes a swirler having axial symmetry about the combustion axis of the combustion chamber for conveying inwardly flowing air, a fuel injector for injecting fuel into the radially inwardly flowing air in such a manner that the air and fuel mix to form an air-fuel mixture having a specified air-fuel ratio and a fuel supply regulator for delivering fuel at a specified rate of fuel delivery. The system further includes a blower for delivering air to the burner at a specified air flow rate so as to produce the specified air-fuel ratio, a swirler air temperature sensor for measuring the temperature of the air delivered to the combustion chamber, and a controller for governing the rate of air delivery based at least on the temperature of the air delivered to the combustion chamber.

In a further embodiment, the system includes a heater head temperature sensor for measuring the temperature of the heater head and a controller for governing the rate of fuel delivery based at least upon the temperature of the heater head and the temperature of the air delivered to the combustion chamber. In yet another embodiment, the system further includes a gas composition sensor for monitoring a gas concentration in the exhaust product of the burner and a controller for governing the rate air delivery based at least upon the temperature of the air delivered to the combustion chamber and the gas composition in the exhaust gas product. The system may also include a flow sensor for measuring the rate of fuel delivery where the controller includes a controller based at least on he temperature of the air delivered to the combustion chamber and the measured rate of fuel delivery.

Brief Description of the Drawings

The invention will be more readily understood by reference to the following description, taken with the accompanying drawings, in which:

FIGS. **1a-1e** depict the principle of operation of a prior art Stirling cycle machine;

FIG. **2** is a side view in cross section of a Stirling cycle engine in accordance with an embodiment of the present invention;

FIG. **3** is a cross-sectional view of a folded guide link drive mechanism for a two-piston machine such as a Stirling cycle machine in accordance with a preferred embodiment of the invention.

FIG. **4** is a perspective view of one embodiment of the folded guide link drive mechanism of Fig. 3.

FIG. **5a** is a cross-sectional view of a Stirling cycle engine employing a pin heat exchanger, in accordance with an embodiment of the present invention;

FIG. **5b** is a magnified perspective detail view of the pin heat exchanger of FIG. 5a;

FIG. **5c** shows a cross-sectional view of the heater head assembly of Fig. 5a with heat transfer pins shown schematically, not to scale, in accordance with a preferred embodiment of the invention.

FIG. **6a** is a perspective view from the bottom of the Stirling cycle engine of Fig. 2, showing branching ducts for enhancing flow uniformity in accordance with an embodiment of the present invention;

FIG. **6b** is a planar view of the system of branching ducts of FIG. 9a;

FIG. **7a** shows a cross-sectional view from the side of a fuel intake manifold for a Stirling cycle engine in accordance with a preferred embodiment of the invention;

FIG.**7b** shows a cross sectional view from the top of the fuel intake manifold of Fig. 10a taken through cut BB;

FIG.**7c** is a cross sectional view from the top of the fuel intake manifold of Fig. 10a taken through cut AA, showing the fuel jet nozzles.;

FIG. **8** is a cross section of a burner and heater head assembly showing the placement of temperature sensors in accordance with an embodiment of the invention.

FIG. **9** shows the relationship of the optimal fuel-air ratio to the air preheat temperature for propane in accordance with a preferred embodiment of the invention.

Detailed Description of Preferred Embodiments

Referring now to FIG. 2, one embodiment of a Stirling cycle engine is shown in cross-section and is designated generally by numeral **28**. While the invention will be described generally with reference to the Stirling engine shown in FIG. 2, it is to be understood that many engines as well as refrigerators may similarly benefit from various embodiments and improvements which are subjects of the present invention. The configuration of Stirling engine **28** shown in FIG. 2 is referred to as an alpha configuration, characterized in that compression piston **30** and expansion piston **32** undergo linear motion within respective and distinct cylinders: compression piston **30** in compression cylinder **34** and expansion piston **32** in expansion cylinder **36**. The alpha configuration is discussed by way of example only, and without limitation of the scope of any appended claims.

In addition to compression piston **30** and expansion piston **32**, the main components of Stirling engine **28** include heater **64**, regenerator **66**, and cooler **68**. Compression piston **30** and expansion piston **32**, referred to collectively as pistons, are constrained to move in reciprocating linear motion within respective volumes **38** and **40**. A cylinder liner **42** may line the respective cylinder surfaces. The volumes of the cylinder interior proximate to the heater **64** and cooler **68** will be referred to, herein, as hot and cold sections, respectively, of engine **28**. The relative phase (the "phase angle") of the reciprocating linear motion of compression piston **30** and expansion piston **32** is governed by their respective coupling to drive mechanism **44** housed in crankcase **46**. Drive mechanism **44**, discussed in greater detail below, may be employed to govern the relative timing of pistons and to interconvert linear and rotary motion. Compression piston **30** and expansion piston **32** are coupled, respectively, to drive mechanism **44** via a first connecting rod **48** and a second connecting rod **50**. The volume **38** of compression cylinder **34** is coupled to cooler **68** via duct **45** to allow cyclic cooling of working fluid. Duct **45**, more particularly, couples compression volume **38** to the annular heat exchangers comprising cooler **68**, regenerator **66**, and heater **64**. Branching of flow between duct **45** and annular plenum **47** is discussed below with reference to Fig. 6.

The operation of drive mechanism **44** is now discussed with reference to FIGS. 3 and 4. Figure 3 is a cross-sectional view of a dual folded guide link drive mechanism designated generally by numeral **300**. The drive mechanism **300** in FIG. 3 comprises two folded guide links **303** and **313**. Pistons **301** and **311** are the displacer and compression pistons, respectively, of a Stirling cycle engine such as described above with respect to FIG. 2. As used in this description and the following claims, a displacer piston is either a piston without a seal or a piston with a seal (commonly known as an "expansion" piston). Displacer piston **301** is rigidly coupled to the piston end of guide link **303** at a piston connection point **302**. Guide link **303** is rotatably connected to a connecting rod **306** at a rod connection point **305**. The piston connection point **302** and the rod connection point **305** define the longitudinal axis **324** of guide link **303**.

Connecting rod **306** is rotatably connected to a crankshaft **308** at a crankshaft connection point **307** which is offset a fixed distance from the crankshaft axis of rotation **326**. The crankshaft axis of rotation **326** is orthogonal to the longitudinal axis **324** of the guide link **303** and the crankshaft axis of rotation **326** is disposed between the rod connection point **305** and the piston connection point **302**. In a preferred embodiment, the crankshaft axis of rotation **326** intersects the longitudinal axis **324**.

An end **328** of guide link **303** is constrained between a pair of rollers **304**. In a preferred embodiment, one of the rollers **304** is spring loaded to maintain rolling contact with the guide link **303**. Alignment of the longitudinal axis **324** of the guide link **303** with respect to piston cylinder **322** is maintained by the rollers **304** and by the piston **301**. As crankshaft **308** rotates about the crankshaft axis of rotation **326**, the rod connection point **305** traces a linear path along the longitudinal axis **324** of the guide link **303**.

Piston **301** and guide link **303** form a lever with the piston **301** at one end of the lever and the rod end **328** of the guide link **303** at the other end of the lever. The fulcrum of the lever is on the line defined by the centers of the rollers **304**. The lever is loaded by a force applied at the rod connection point **305**. As rod connection point **305** traces a path along the longitudinal axis of the guide link **303**, the distance between the rod connection point **305** and the fulcrum, the first lever arm, will vary from zero to one-half the stroke distance of the piston **301**. The second lever arm is the distance from the fulcrum to the piston **301**. The lever ratio of the second lever arm to the first lever arm will always be greater than one, preferably in the range from 5 to 15. The lateral force at the piston **301** will be the forced applied at the rod connection point **305** scaled by the lever ratio; the larger the lever ratio, the smaller the lateral force at the piston **301**.

The compression piston **311** is rigidly coupled to the piston end of guide link **313** at a piston connection point **312**. Guide link **313** is rotatably connected to a connecting rod **316** at a rod connection point **315**. The piston connection point **312** and the rod connection point **315** define the longitudinal axis of guide link **313**. Connecting rod **316** is rotatably connected to the crankshaft **308** at a crankshaft connection point **317** which is offset a fixed distance from the crankshaft axis of rotation **326**. An end **330** of guide link **313** is constrained between a pair of rollers **314**. As discussed above, in a preferred embodiment on of the rollers **314** is spring loaded to maintain rolling contact with the guide link **313**. The operation of guide link **313** is similar to that described above with respect to guide link **303**. Alignment of the longitudinal axis of guide link **313** with respect to piston cylinder320 is maintained by the rollers **314** and by the piston **301**. As crankshaft **308** rotates about the crankshaft axis of rotation **326**, the rod connection point **305** traces a linear path along the longitudinal axis of the guide link **313**.

Figure 4 is a perspective view of the dual folded guide link drive mechanism shown in Figure 3. Compression piston **311** and displacer piston **301** undergo linear motion within respective and distinct cylinders: compression piston **311** in compression cylinder **320** and displacer piston **301** in expansion cylinder **322**. Guide link **303** and guide link **313** are rigidly coupled to displacer piston **301** and compression piston **311** at piston connection points **302** and **312** respectively (shown in Figure 3). Connecting rods **306** and **316** are rotationally coupled at connection points **305** and **315** of the distal ends of guide links **303** and **313** to crankshaft **308** at crankshaft connection points **307** and **317** (shown in Figure 3). Lateral loads on guide links **303** and **313** are taken up by roller pairs **304** and **314**.

Referring now to FIGS. 5a-5c, a novel structure is depicted, in accordance with an embodiment of the present invention, for transferring large amounts of heat from the combustion source to the interior of Stirling cycle engine **28**, shown in cross section. In order to increase the efficiency of heat transfer from hot gases **300**, generated by burner **150**, to the working fluid contained in the interior volume **306** of the engine, a large wetted surface area, on either side of heater head **64** is required. To achieve the high surface area, a large number of metal pins **310** are fabricated on either one of or both the interior surface **312** and exterior surface **314** of heater head **64**. Fabrication may be accomplished at low cost, such as by investment casting. Metal pins **310** not only increase the wetted surface area on either side of heater head **64** but also create turbulent wakes that increase fluid mixing and thereby further increase the flow of heat. This structure may also be employed for heat transfer at the cooler **68** (shown in FIG. 2) or in any application where efficient heat transfer is required between volumes of gases. Figure 5c shows a cross-sectional view of the heater head assembly of Figure 5a with het transfer pins **130** and **124** shown schematically in accordance with a preferred embodiment of the invention. In Figure 5c, inner heat transfer pins **124** and outer heat transfer pins **130** are located along the sides of the heater head **64**.

Referring to FIG. 6a, a perspective view is shown of a system of header ducts **400** providing for the flow of working fluid between compression volume **38** and the annular region of fluid flow through the heat exchange network, namely past cooler head **68**, through regenerator **66** (shown in FIG. 2), and past heater head **64** (shown in FIG. 2). The annular flow of working fluid culminates at annular header **47** to which branching ducts **400** are coupled for creating equal-length flow passages between cylinder volume **38** and the entire annular region of header **47**. By substantially equalizing the flow impedance between every portion of the annular flow region and the cylinder volume, losses due to flow non-uniformities through the heat exchangers may be advantageously reduced, and, additionally, the flow of working fluid within a loop confined to the heat exchange region and thereby lost for purposes of mechanical work may be minimized. FIG. 6b shows a schematic of the system of branching ducts **400** of FIG. 6a, "unwrapped" into a planar view, showing the fluid communication via branching ducts **400** between compression space **38** and annular header

47.

While Stirling engines are capable of providing high thermal efficiency and low emission of pollutants, these objectives impose requirements of thermal efficiency, in particular, on a burner **806** employed to heat heater head **808** of the Stirling engine as shown in Figure 8. Components of such thermal efficiency include the efficient pumping of oxidant (typically, air, and, referred to herein and in any appended claims, without limitation, as "air") through the burner **806** to provide combustion, and the recovery of hot exhaust leaving the heater head **808**. In many applications, air (or other oxidant) is pre-heated, prior to combustion, nearly to the temperature of the heater head **808**, so as to achieve the stated objectives of thermal efficiency. There is still a considerable amount of energy left in the combustion gases after the heater head of the Stirling engine has been heated, and, as known to persons skilled in the art, a heat exchanger may be used to transfer heat from the exhaust gases to the combustion air prior to introduction into the burner. In order to achieve high efficiency and low emissions, the burner must provide substantially complete combustion. In order to achieve substantially complete combustion, a measured amount of air as well as a clean burning fuel, preferably propane, are delivered to the burner. The fuel and air flow rates are controlled in order to allow for ignition of a flame in the burner as well as for clean emissions after ignition. The fuel and air must also be well-mixed with sufficient amounts of oxygen to limit the emission of carbon monoxide (CO) and hydrocarbons and, additionally, must be burned at low enough flame temperatures to limit the formation of oxides of nitrogen (NO_x).

The high temperature of preheated air, desirable for achieving high thermal efficiency, complicates achieving low-emission goals by making it difficult to premix the fuel and air and requiring large amounts of excess air in order to limit the flame temperature. As used herein and in any appended claims, the term "auto-ignition temperature" is defined as the temperature at which a fuel will ignite without a temperature-decreasing catalyst under existing conditions of air and fuel pressure. The typical preheated air temperature exceeds the auto-ignition temperature of most fuels, potentially causing the fuel-air mixture to ignite before entering the combustion chamber. One solution to this problem is to use a non-premixed diffusion flame. However, since such diffusion flames are not well-mixed, higher than desirable emissions of CO, HC and NO_x result. A detailed discussion of flame dynamics is provided by Turns, *An Introduction to Combustion: Concepts and Applications*, (McGraw-Hill, 1996), which is incorporated herein by reference. Any increased air flow provided to limit flame temperatures typically increases the power consumed by an air pump or blower, thereby degrading overall engine efficiency.

In accordance with the present invention, low emissions and high efficiency may be provided by producing a pre-mixed flame even in the presence of air heated above the auto-ignition temperature of the fuel, and, additionally, by minimizing the pressure drop between the air inlet and the flame region, thereby minimizing blower power consumption.

The term "flame speed" is defined as the speed at which a flame front will propagate through a particular fuel-air mixture. Within the specification and the following claims, the term "combustion axis" shall refer to the direction of predominant fluid flow upon combustion of the fluid.

Referring now to FIGS. 7a-7c, an intake manifold **699** is shown for application to a Stirling cycle engine or other combustion application in accordance with an embodiment of the present invention. In accordance with a preferred embodiment of the invention, fuel is pre-mixed with air that may be heated above the fuel's auto-ignition temperature and a flame is prevented from forming until the fuel and air are well-mixed and in the combustion chamber **809** (shown in Figure 8. FIG. 7a shows a preferred embodiment of the apparatus including an intake manifold **699** and a combustion chamber **710**. The intake manifold **699** has an axisymmetrical conduit **701** with an inlet manifold **703** for receiving air **700** supplied via the blower **728**. Air **700** is pre-heated to a temperature, typically above 1000 K, which may be above the auto-ignition temperature of the fuel. Conduit **701** conveys air **700** flowing inward radially with respect to combustion axis **720** to a swirler **702** disposed within the conduit **701**.

Fig. 7b shows a cross sectional view of the conduit **701** including swirler **702** in accordance with an embodiment of the invention. In the embodiment of FIG. 7b, swirler **702** has several spiral-shaped vanes **730** for directing the flow of air **700** radially inward and imparting a rotational component on the air. The diameter of the swirler section of the conduit decreases from the inlet **732** to the outlet **734** of swirler **702** as defined by the length of swirler vanes **730**. The decrease in diameter of swirler vanes **730** increases the flow rate of air **700** in substantially inverse proportion to the diameter. The flow rate is increased so that it is above the flame speed of the fuel. At outlet **734** of swirler **702**, fuel **706**, which in a preferred embodiment is propane, is injected into the inwardly flowing air.

In a preferred embodiment, fuel **706** is injected by fuel injector **704** through a series of nozzles **736** as shown in FIG. 7c. More particularly, FIG. 7c shows a cross sectional view of conduit **701** and includes the fuel jet nozzles **736**. Each of the nozzles **736** is positioned at the exit of the swirler vanes **730** and is centralized between two adjacent vanes. Nozzles **736** are positioned in this way for increasing the efficiency of mixing the air and fuel. The fuel jet nozzles **736** are sized to provide jets of fuel that extend at least half way across the conduit **701** (shown in Figures 7a and 7b). Calculations to size the fuel jet nozzles **736** are well known in the art and described in Boer and Chigier, "Combustion Aerodynamics," John Wiley & Sons, 1972. Nozzles **736** simultaneously inject the fuel **706** across the air flow **700**. Since the air flow is faster than the flame speed, a flame will not form at that point even though the temperature of the air and fuel mixture is above the fuel's auto-ignition temperature. In a preferred embodiment, where propane is used, the preheat temperature, as governed by the temperature of the heater head, is approximately 900 K.

Referring again to FIG. 7a, the air and fuel, now mixed, referred to hereafter as "air-fuel mixture" **709**, is transitioned in direction through a throat **708** which has a contoured fairing **722** and is attached to the outlet **707** of the conduit **701**. Fuel **706** is supplied via fuel regulator **724**. Throat **708** has an inner radius **714** and an outer dimension **716**. The transition of the air-fuel mixture is from a direction which is substantially transverse and radially inward with respect to combustion axis **720** to a direction which is substantially parallel to the combustion axis. The contour of the fairing **722** of throat **708** has the shape of an inverted bell such that the cross sectional area of throat **708** with respect to the combustion axis remains constant from the inlet **711** of the throat to outlet **712** of the throat. The contour is smooth without steps and maintains the flow speed from the outlet of the swirler to the outlet of the throat **708** to avoid separation and the resulting recirculation along any of the surfaces. The constant cross sectional area allows the air and fuel to continue to mix without decreasing the flow speed and causing a pressure drop. A smooth and constant cross section produces an efficient swirler, where swirler efficiency refers to the fraction of static pressure drop across the swirler that is converted to swirling flow dynamic pressure. Swirl efficiencies of better than 80% may typically be achieved by practice of the invention. Thus, the parasitic power drain of the combustion air fan may be minimized.

Outlet **712** of the throat flares outward allowing the air-fuel mixture **709** to disperse into the chamber **710** slowing the air-fuel mixture **709** thereby localizing and containing the flame and causing a toroidal flame to form. The rotational momentum generated by the swirler **602** produces a flame stabilizing ring vortex as well known in the art.

As discussed above, the fuel and air flow rates are controlled in order to allow for ignition of a flame in the burner as well as for clean emissions after ignition. Referring to Figure 7a, burner controller **726** is used to control the fuel and air flow rates provided by fuel regulator **724** and blower **728** respectively. The fuel regulator **724** is set to an initial value for ignition. Once the flame is proved, the burner controller **726** varies the fuel flow rate to control the heater head temperature as measured by a head temperature sensor **804** (shown in Figure 8). A flame is proved when a flame detector detects the presence of the flame. There are several types of flame detectors including thermocouples and ultraviolet sensors known in the art.

The output (or air mass flow rate) of the combustion air blower **728** is set by the burner controller **726** to control the fuel-air ratio in the combustion chamber **809** (shown in Fig. 8). The fuel-air ratio is the ratio of the fuel mass flow rate over the air mass flow rate and is the primary factor affecting emissions. The blower **728** controls the fuel-air ratio by increasing or decreasing the air mass flow rate relative to the fuel mass flow rate. For example, in order to hold the fuel-air ratio constant, the burner controller **726** will increase the blower output as the fuel regulator **724** increases its output and vice versa. The desired fuel-air ratio and the fuel flow rate may be changing at the same time, so the burner controller **726** will change the output of the blower **728** to accommodate both the change in desired fuel-air ratio and the fuel flow rate.

Minimizing the emissions of carbon monoxide (CO), hydrocarbons (HC) and oxides of nitrogen (N_{ox}) requires a lean fuel-air mixture which still achieves complete combustion. A lean fuel-air mixture has more air than a stoichiometric mixture (i.e., 15.67 grams of air per gram of propane, for example). As more air is added to a fixed amount of fuel, the emissions of CO, HC and NO_x will decrease until the amount of air in the fuel-air mixture is large enough that the flame becomes unstable. At this point, pockets of the fuel-air mixture will pass through the burner without complete combustion. Incomplete combustion of the fuel-air mixture produces large amounts of CO and HC. The CO and HC emissions will quickly increase as more air is added to the fuel-air mixture until the flame extinguishes at a Lean Blow-Out limit ("LBO"). The LBO will increase as the temperature of the incoming air (i.e, the preheated air) increases. As a result, the optimal fuel-air ratio of the fuel-air mixture decreases as the temperature of the preheated air increases during the warmup phase of the Stirling engine. Once the engine is warmed up, the fuel-air ratio is held constant.

Accordingly, the fuel-air ratio must first be controlled to provided the optimal fuel-air ratio for ignition. Once the flame is proved, the fuel-air ratio is controlled to minimize emissions based upon the temperature of the preheated air and the fuel type. When the fuel flow rate is increased or decreased to adjust the temperature of the heater head, the air flow rate is also adjusted to maintain the desired fuel-air ratio.

A given fuel will only ignite over a limited range of fuel-air ratios. At ignition, an ignition fuel-air ratio is chosen which is equal to or less than the stoichiometric fuel-air ratio corresponding to the fuel being used. In a preferred embodiment, where the fuel is propane, the ignition fuel-air ratio is set to 0.1 grams propane per gram of air. The ignition fuel-air ratio is maintained until the flame stabilizes and temperature of the interior of the combustion chamber increases to a warmup temperature. Referring to Figure 8, the temperature of the combustion chamber **809** is typically determined by measuring the temperature of the heater head **808** or by allowing a predetermined time interval for the combustion chamber to heat. A temperature sensor, such as thermocouple **804**, may be used to measure the temperature of heater head **808**. In a preferred embodiment, the ignition fuel-air ratio is held until the heater head temperature reaches 300°C and the flame has been lit for 5 seconds.

Once the flame is stabilized, and the temperature of the combustion chamber **809** reaches the desired warmup temperature, the fuel-air ratio is then controlled based upon the air preheat temperature and the fuel type. As described above, the optimal fuel-air ratio **901** of the fuel-air mixture decreases as the temperature of the preheated air **903** increases as shown in Figure 9. The temperature of the preheated air is measured using a temperature sensor, such as a thermocouple **810**, in an air swirler **802** coupled to the combustion chamber **806** as shown in Figure 8. The air preheat temperature can also be inferred from the heater head **808** temperature by subtracting several hundred degrees Celsius from the heater head temperature. In a preferred embodiment, the air preheat temperature is taken as the heater head temperature minus 300°C.

The optimal fuel-air ratio will first decrease linearly with the preheated air temperature from a "start" fuel-air ratio for room temperature air to a "run" fuel air ratio, for a warmed up preheated air temperature. The air is considered fully warmed up when it exceeds the known auto-ignition temperature for the fuel. For example, the auto-ignition temperature for propane is 490°C. In a preferred embodiment, where the fuel is propane, the "start" fuel-air ratio is 0.052 grams of fuel to grams of air, which results in approximately 4% oxygen in the exhaust of the Stirling engine. The "run" fuel-air ratio in the preferred embodiment, is 0.026 grams of fuel to grams of air, which results in approximately 13% oxygen in the exhaust of the Stirling engine.

The fuel-air ratio may be determined by measuring the air and fuel flow rates. A pressure sensor may be used to measure the air-flow rate at the blower **728** (shown in Figure 7a). The fuel flow rate may be determined by measuring the pressure upstream and downstream of a set of fuel control valves of fuel regulator **724** (shown in Figure 7a) and by monitoring which of the valves is currently open. In an alternative embodiment, the fuel-air ratio may be based on a measurement of the oxygen content in the exhaust of the Stirling engine. An oxygen sensor may be placed in the engine to sample the exhaust gas and measure the percentage of oxygen in the exhaust of the engine.

The devices and methods described herein may be applied in other applications besides the Stirling engine in terms of which the invention has been described. The described embodiments of the invention are intended to be merely exemplary and numerous variations and modifications will be apparent to those skilled in the art. All such variations and modifications are intended to be within the scope of the present invention as defined in the appended claims.

WE CLAIM:

1. A method of combusting a fuel and air in a burner of an external combustion engine having a heater head, the fuel and air combined to form a fuel-air mixture characterized by a fuel-air ratio, the fuel-air mixture when combusted producing an exhaust gas product, the method comprising:

igniting the fuel-air mixture to form a flame at a first fuel-air ratio produced by a first air flow rate and a fuel flow rate;

increasing the air flow rate to produce a second fuel-air ratio;

controlling the fuel flow rate based at least on a temperature of the heater head; and

maintaining the flame at the second fuel-air ratio by adjusting the air flow rate based at least on the fuel flow rate.

2. A method according to claim 1, wherein the second fuel-air ratio is maintained by adjusting the air flow rate based at least on an oxygen concentration in the exhaust gas.

3. A method according to claim 1, wherein the second fuel-air ratio is maintained by adjusting the air flow rate based at least on the fuel flow rate and a temperature of the air.

4. A method according to claim 1, wherein the second fuel-air ratio is maintained by adjusting the air flow rate based at least on the temperature of the air and an oxygen concentration in the exhaust gas.

5. A method according to claim 1, wherein igniting the fuel and air, the fuel having an auto-ignition temperature and a flame speed includes propelling the air at a speed above the flame speed into an inlet of a throat, the throat also having an outlet and a constant cross sectional area from inlet to outlet and mixing fuel into the air forming the fuel-air mixture, the fuel-air mixture exiting the outlet, such that a flame is created in the air fuel mixture outside the outlet of the throat.

6. A method according to claim 1, wherein the fuel has an auto-ignition temperature and the fuel-air mixture is ignited at a first air temperature which is less than or equal to the auto-ignition temperature of the fuel.

7. A method according to claim 1, wherein the fuel has an auto-ignition temperature and the flame is maintained at a second air temperature which is greater than the auto-ignition temperature of the fuel.

8. A system for operating a combustion chamber of a thermal cycle engine having a heater head, the combustion chamber characterized by a combustion axis and for delivering heat to the heater head of the engine by combusting a fuel in air to produce heat and an exhaust gas product, the system comprising:

a swirler having axial symmetry about the combustion axis of the combustion chamber for conveying inwardly flowing air;

a fuel injector for injecting fuel into the radially inwardly flowing air in such a manner that the air and fuel mix to form an air-fuel mixture having a specified air-fuel ratio;

a fuel supply regulator for delivering fuel at a specified rate of fuel delivery;

a blower for delivering air to the burner at a specified air flow rate so as to produce the specified air-fuel ratio;

a swirler air temperature sensor for measuring the temperature of the air delivered to the combustion chamber; and

a controller for governing the rate of air delivery based at least on the temperature of the air delivered to the combustion chamber.

9. A system according to claim 8, further including:

a heater head temperature sensor for measuring the temperature of the heater head; and

a controller for governing the rate of fuel delivery based at least upon the temperature of the heater head.

10. A system according to claim 8, further including:

a gas composition sensor for monitoring a gas concentration in the exhaust product of the burner, wherein the controller for governing the rate of air delivery includes a controller based at least upon the temperature of air delivered to the combustion chamber and the gas concentration in the exhaust gas product.

11. A system according to claim 8, further including a flow sensor for measuring the rate for fuel delivery wherein the controller for governing the rate of air delivery includes a controller based at least upon the temperature of the air delivered to the combustion chamber and the measured rate of fuel delivery.

12. A system according to claim 8, wherein the air temperature is estimated based on the temperature of the heater head.

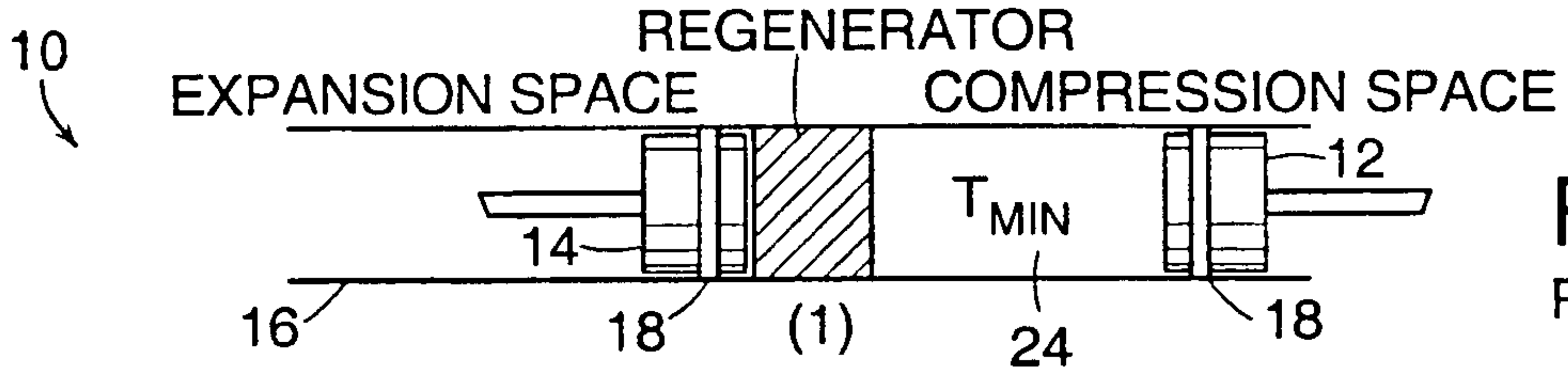


FIG. 1a
PRIOR ART

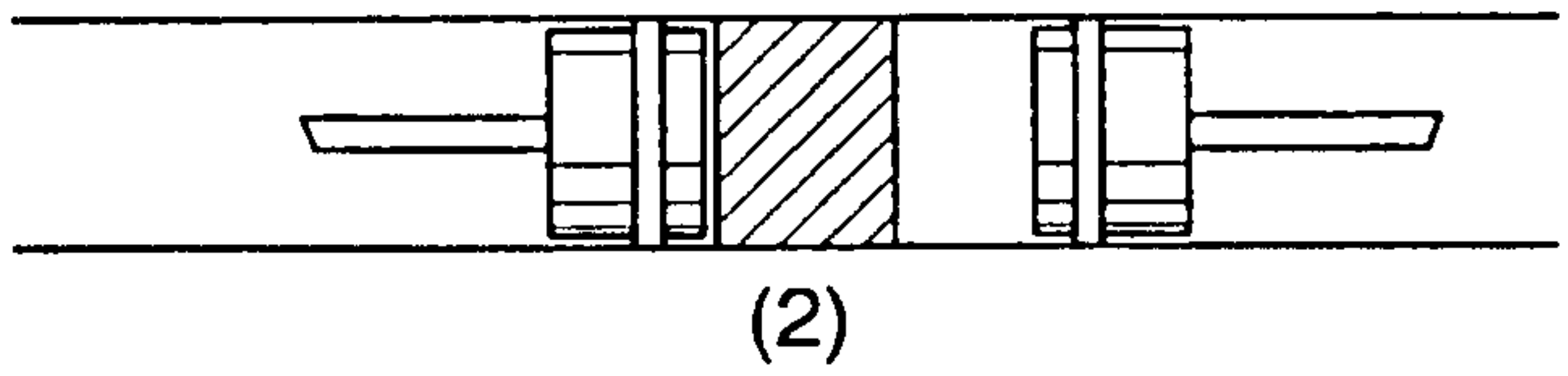


FIG. 1b
PRIOR ART

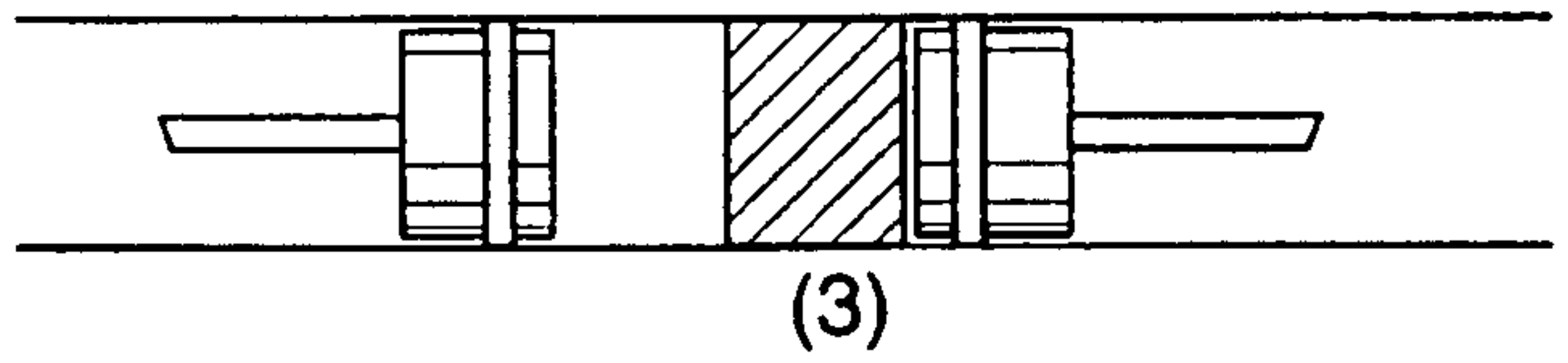


FIG. 1c
PRIOR ART

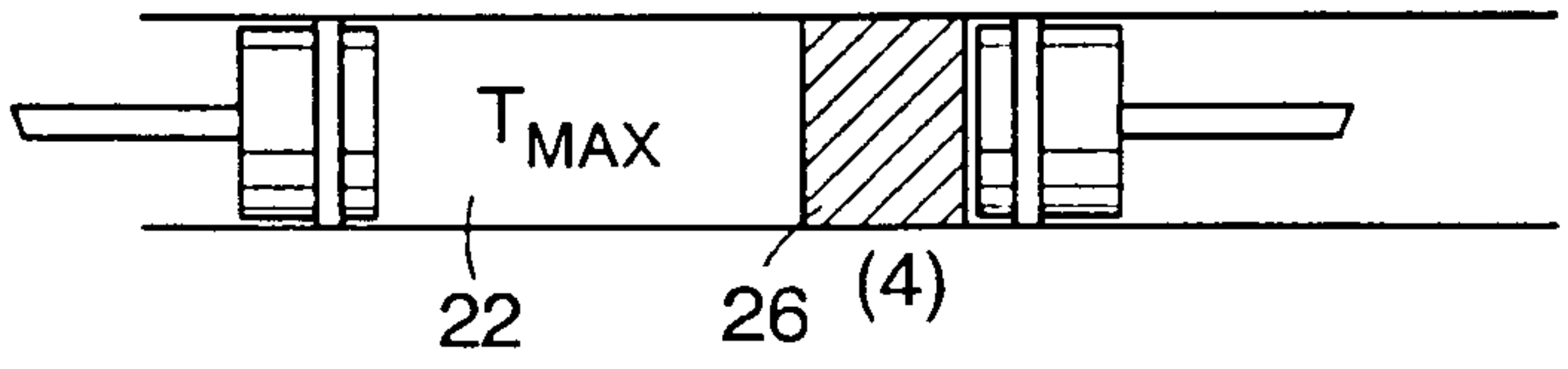


FIG. 1d
PRIOR ART

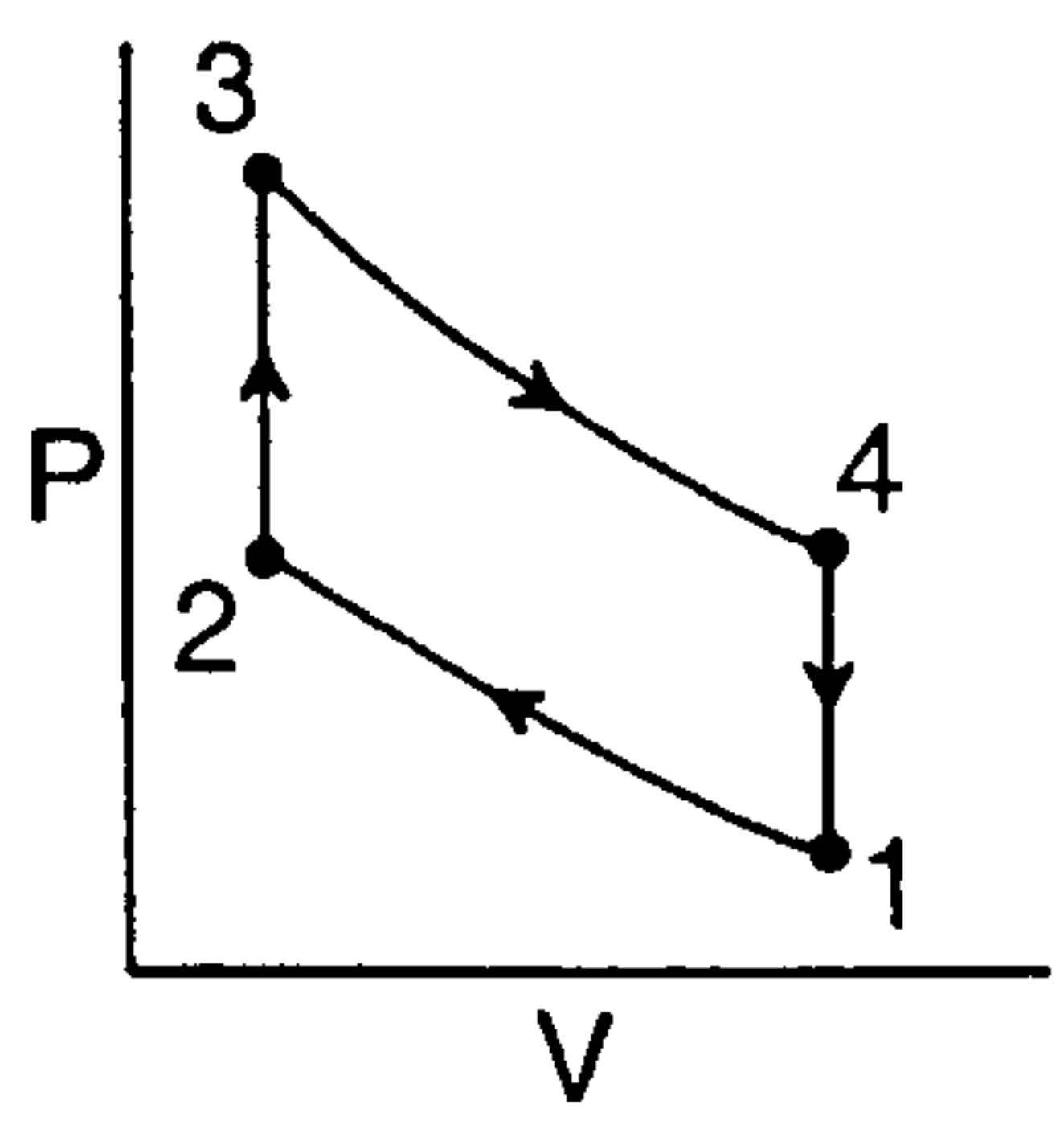


FIG. 1e
PRIOR ART

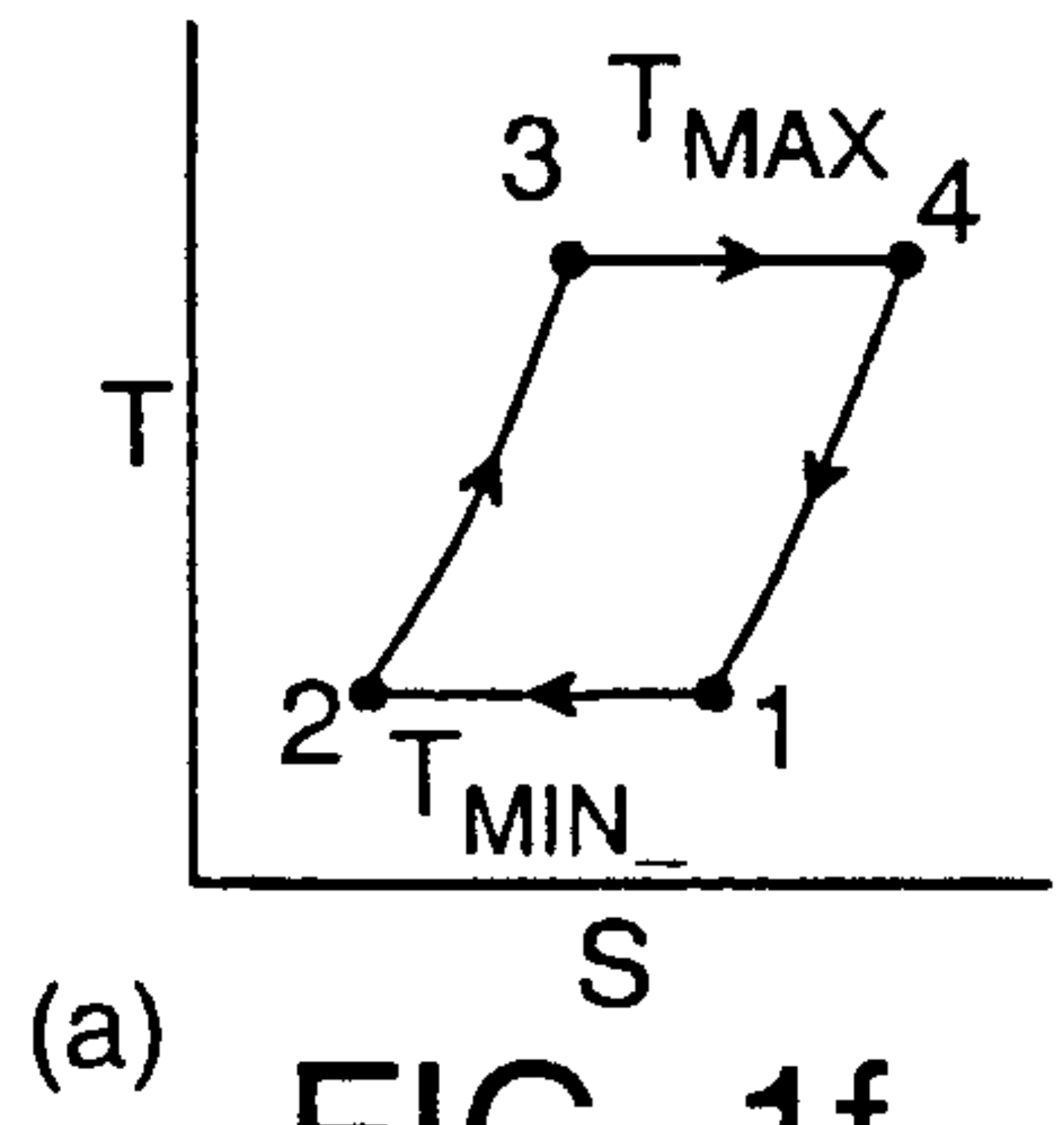


FIG. 1f
PRIOR ART

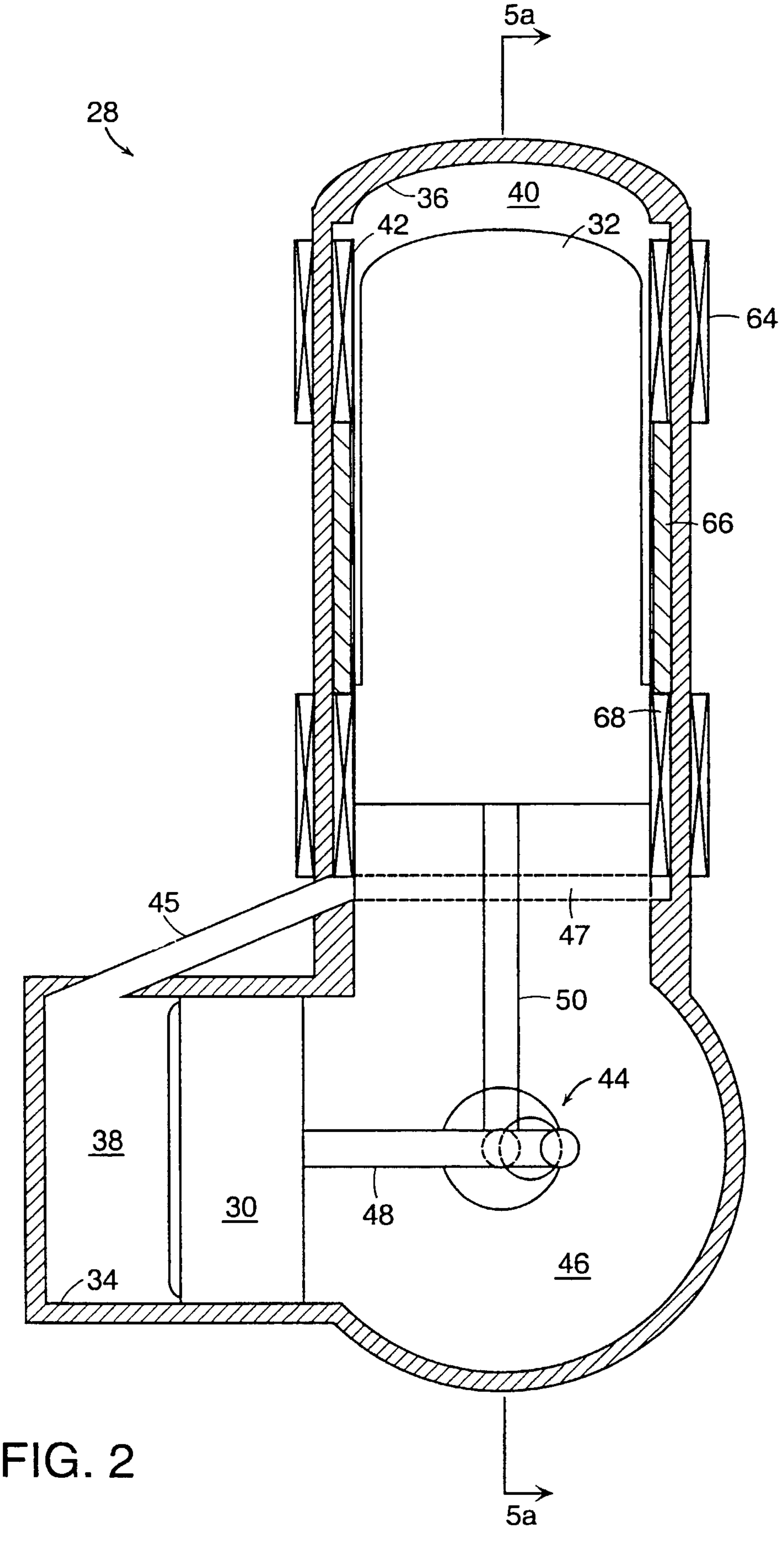


FIG. 2

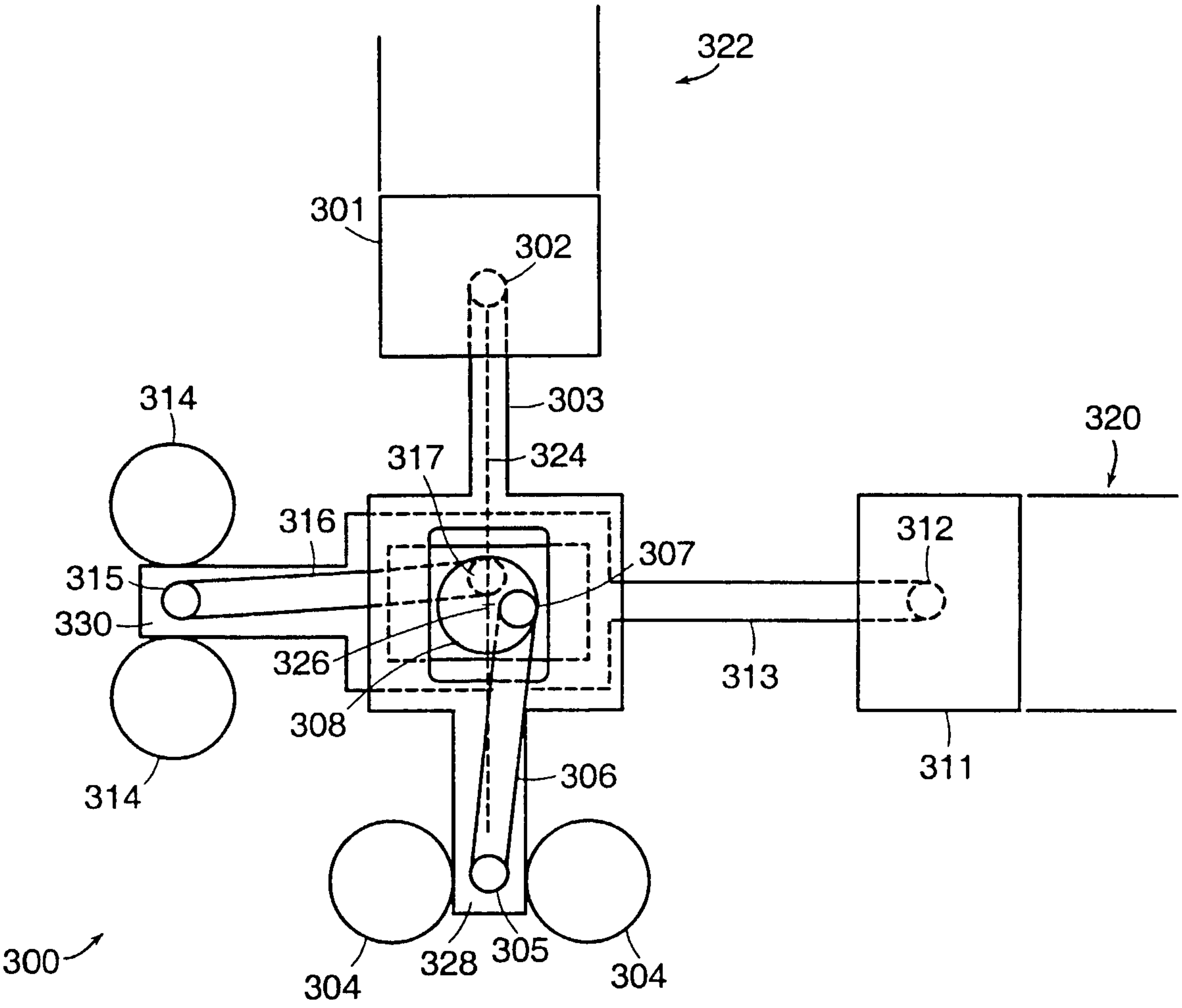

322
301
302
303
324
314
320
317
316
307
312
315
330
326
308
313
311
306
314
300
304
328
305
304

FIG. 3

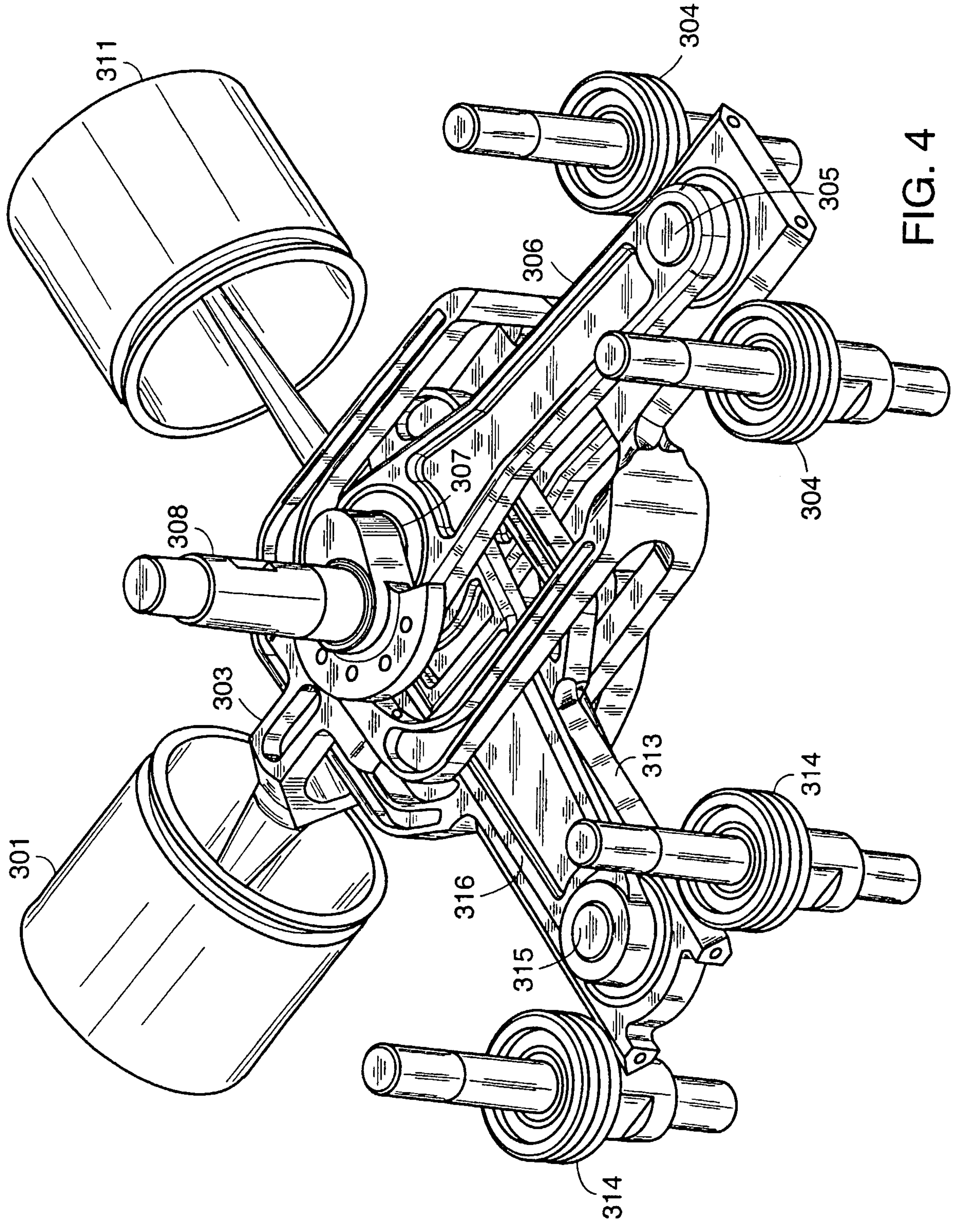


FIG. 4

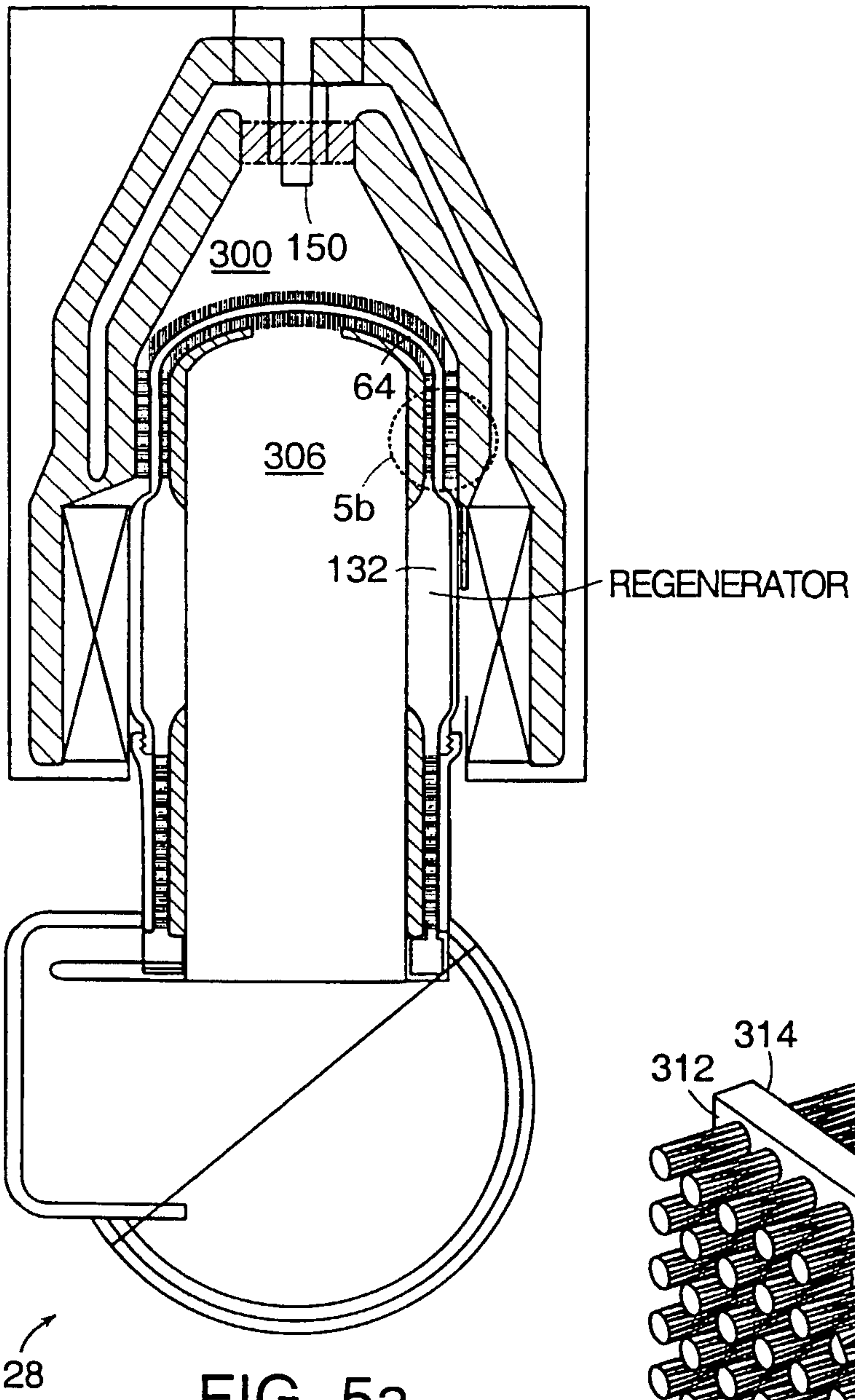


FIG. 5a

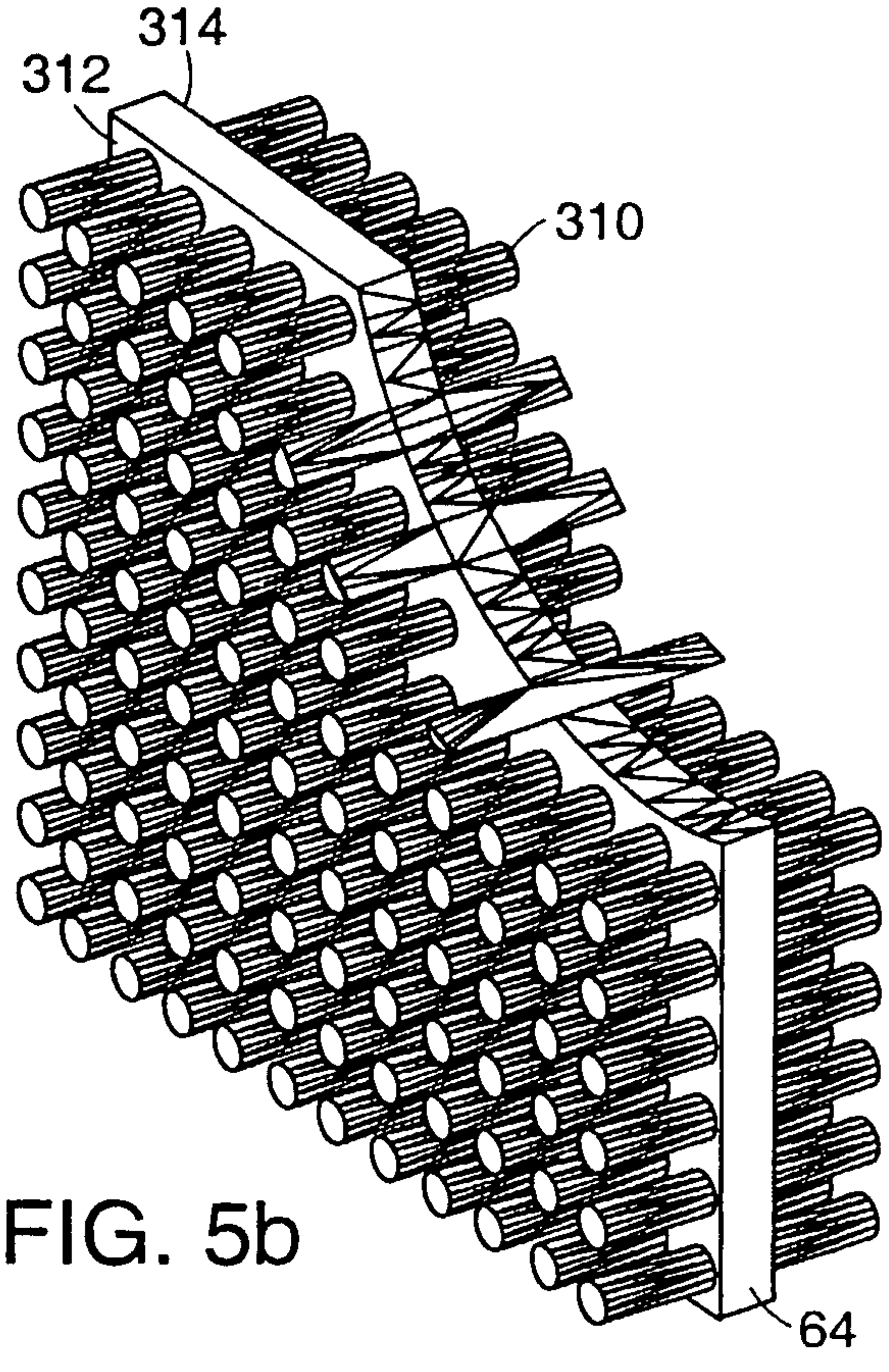


FIG. 5b

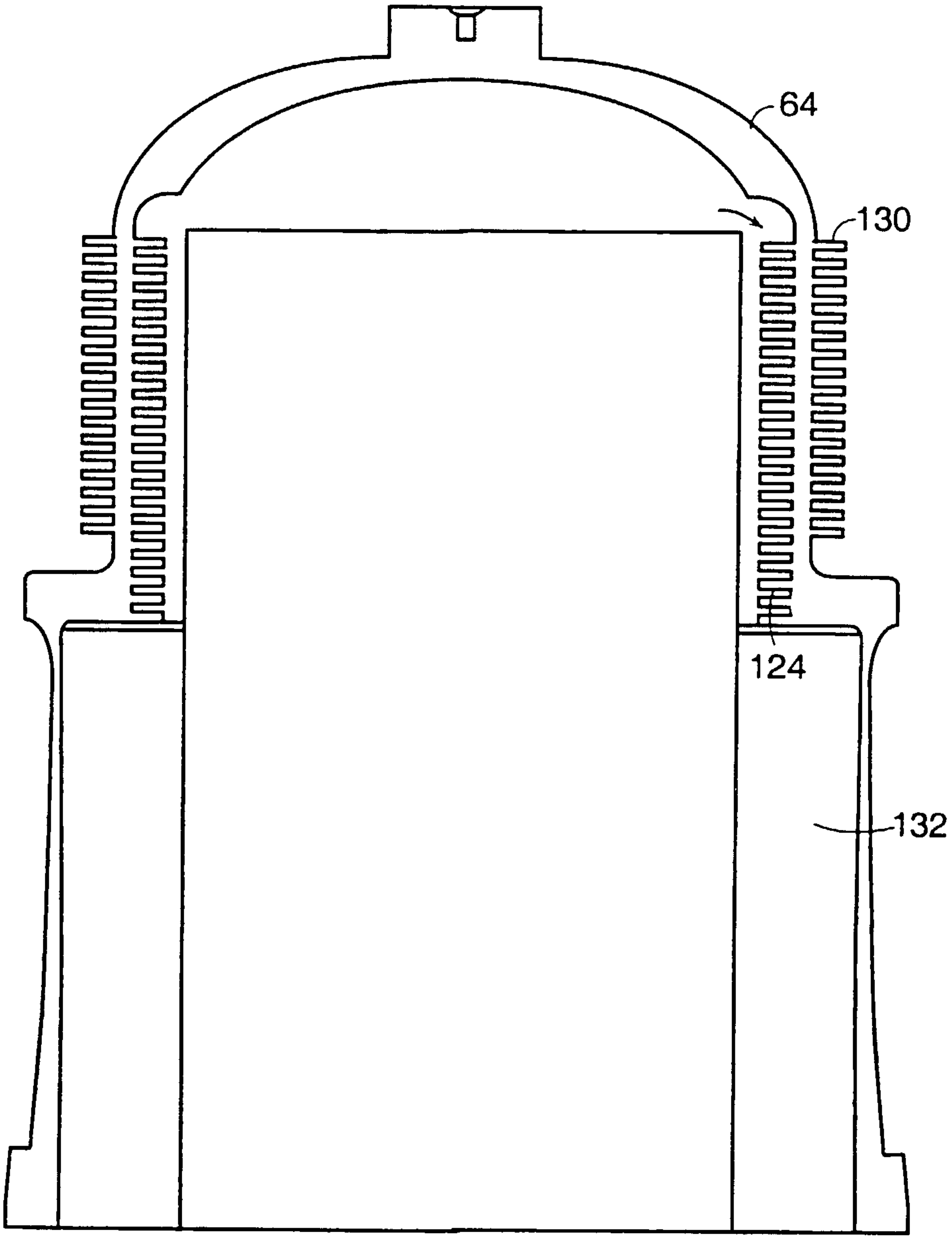


FIG. 5c

"UNWRAPPED" VIEW OF HEADER DUCTS

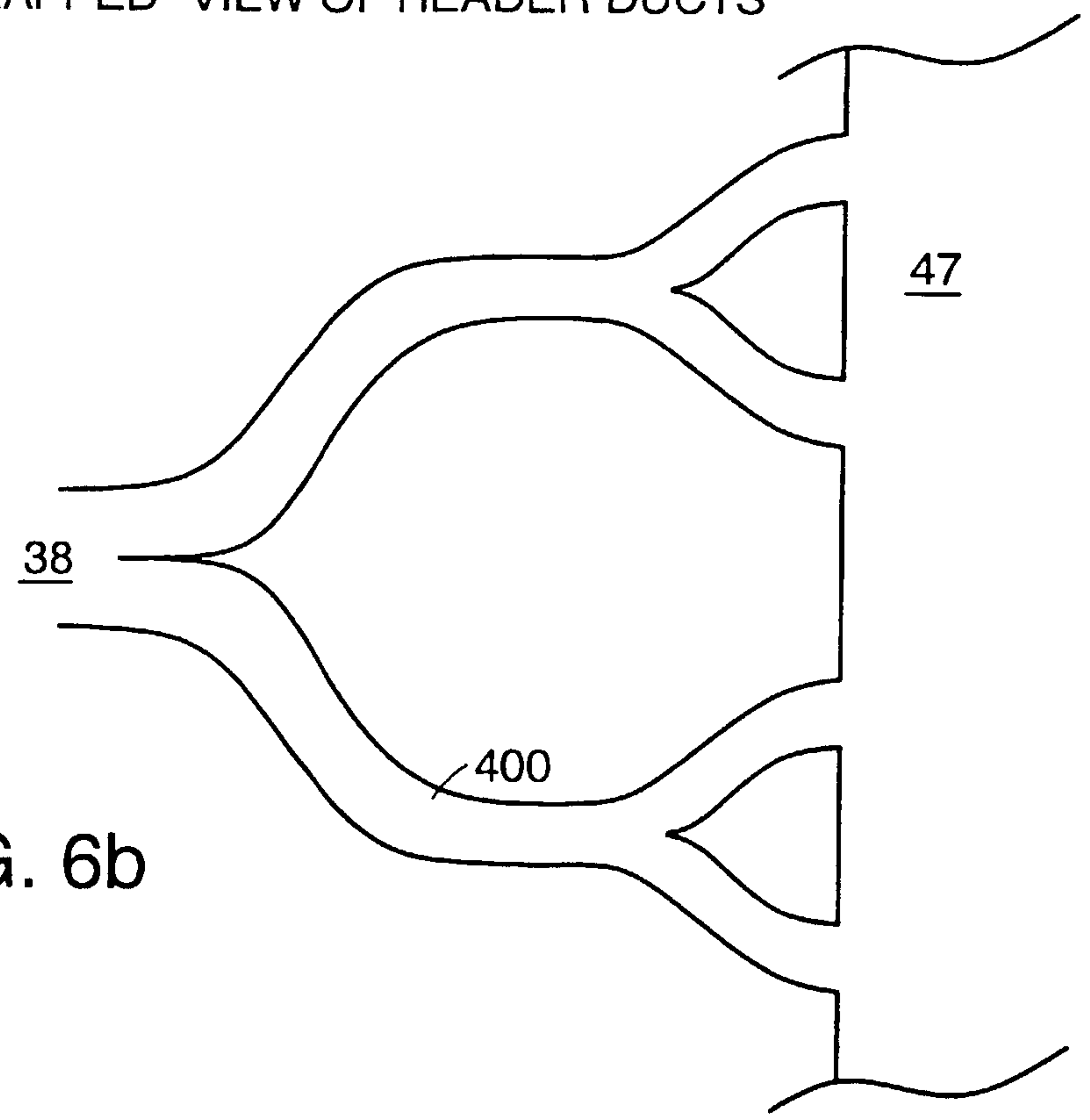


FIG. 6b

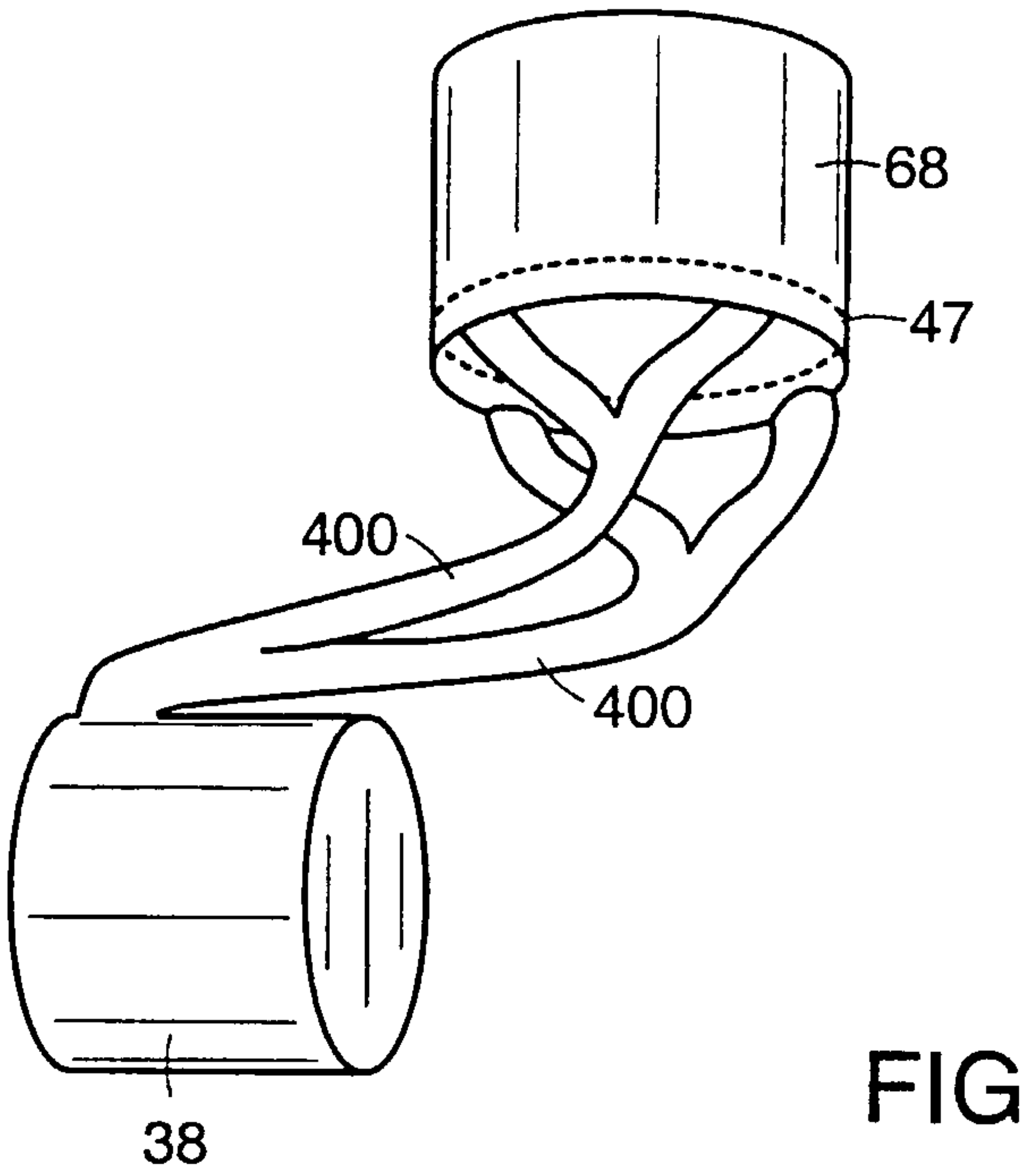


FIG. 6a

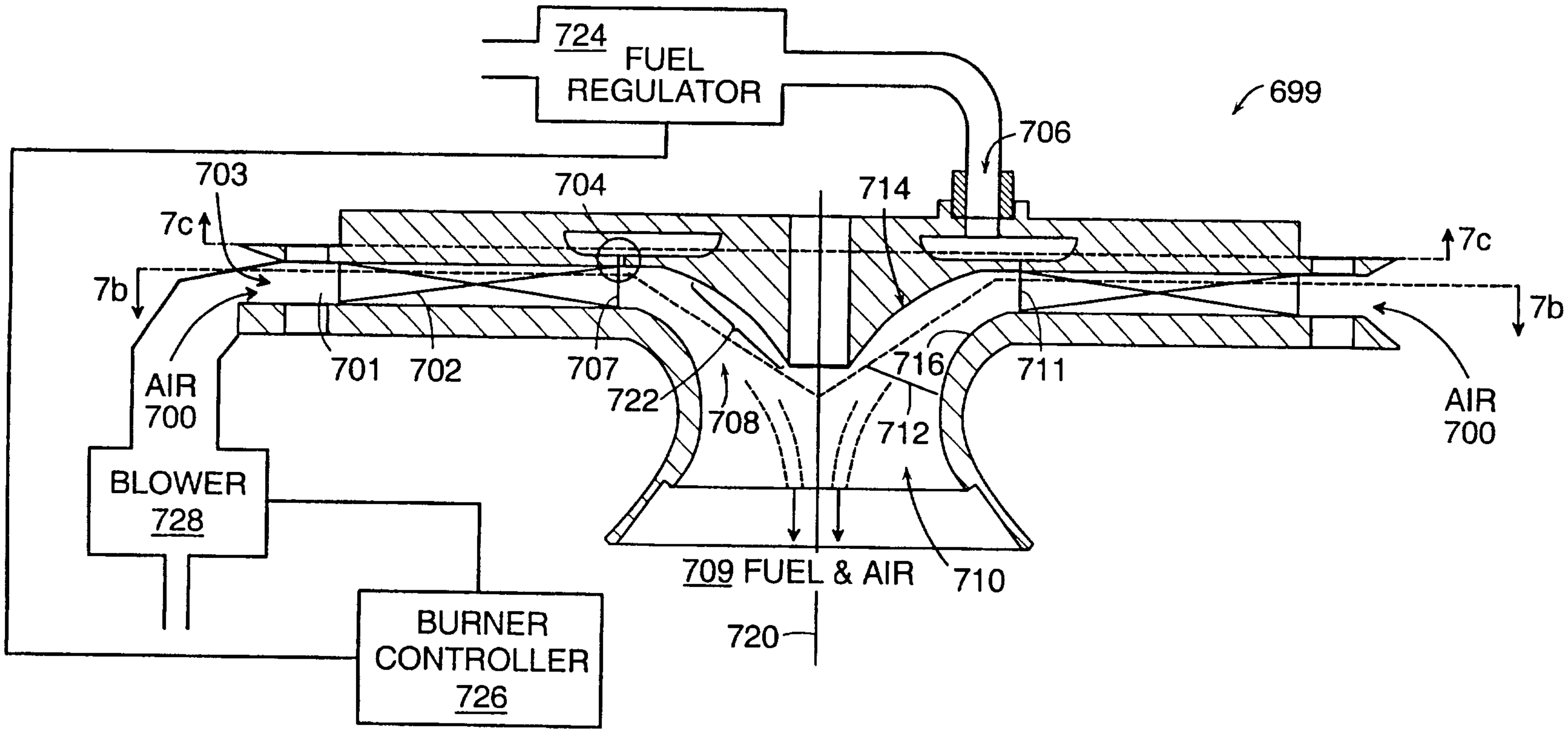

724
FUEL
REGULATOR
699
706
703
704
714
7c
7c
7b
7b
701
702
707
722
708
716
712
711
AIR
700
AIR
700
BLOWER
728
BURNER
CONTROLLER
726
709 FUEL & AIR
710
720

FIG. 7a

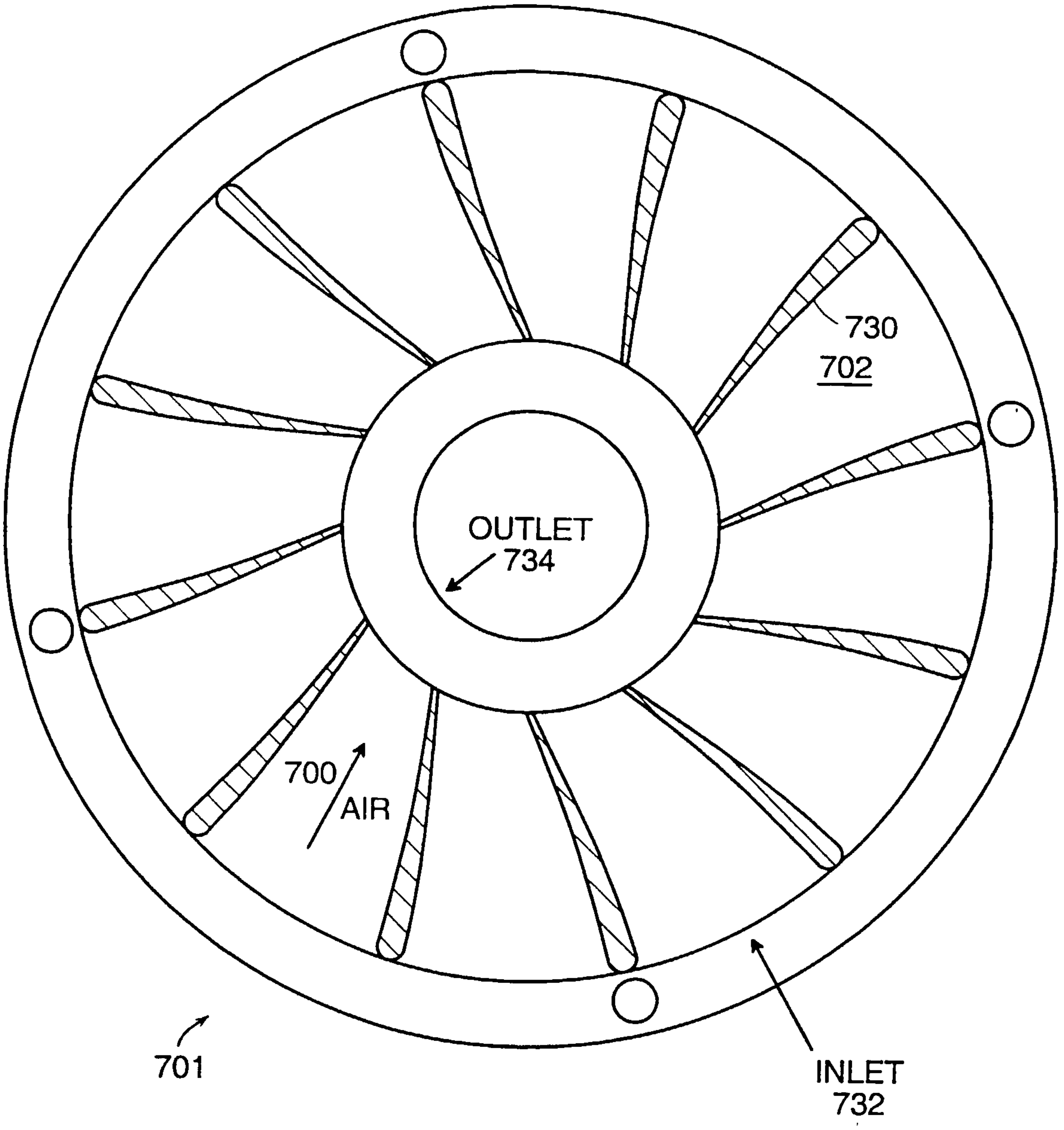


FIG. 7b

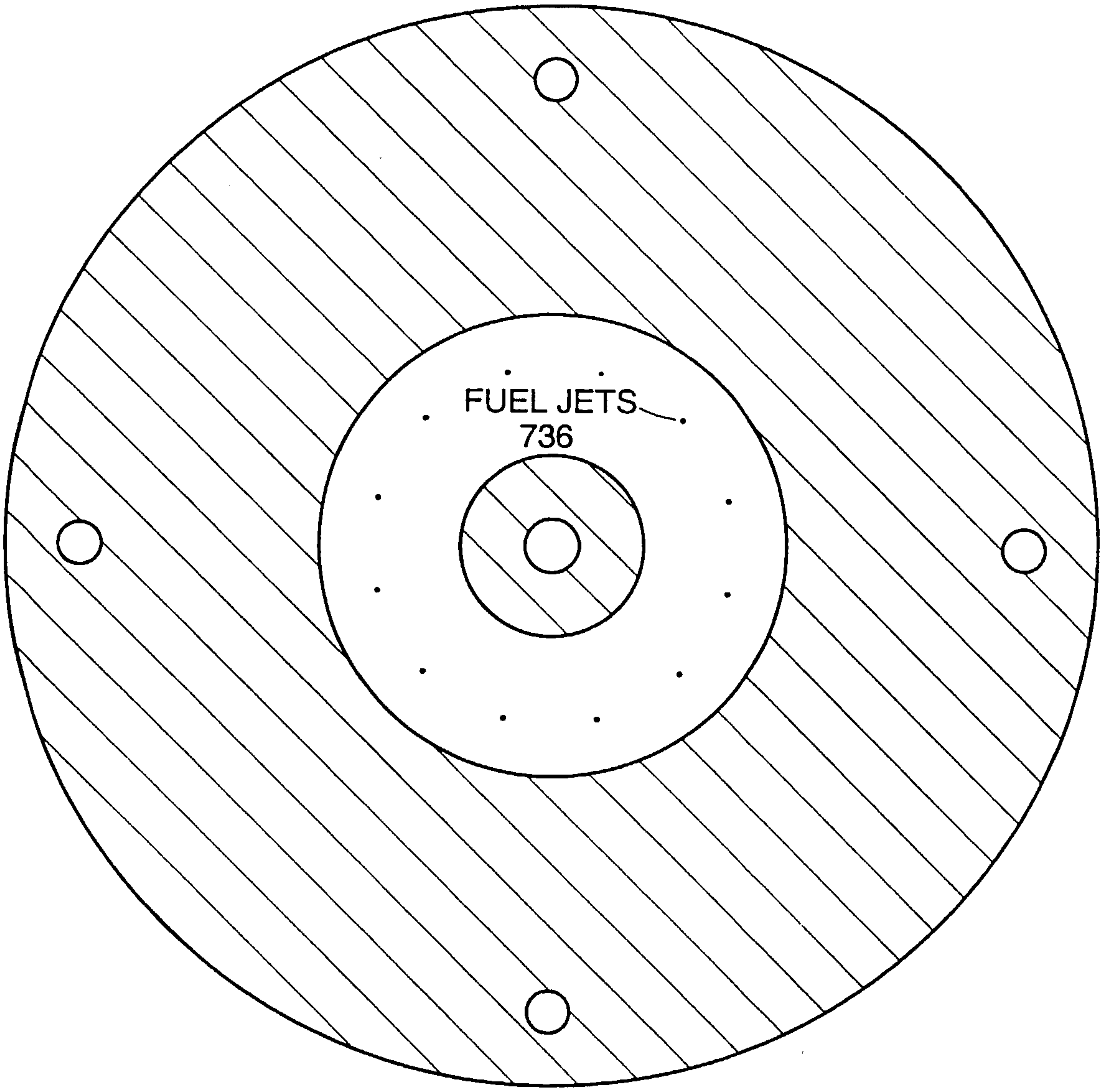


FIG. 7c

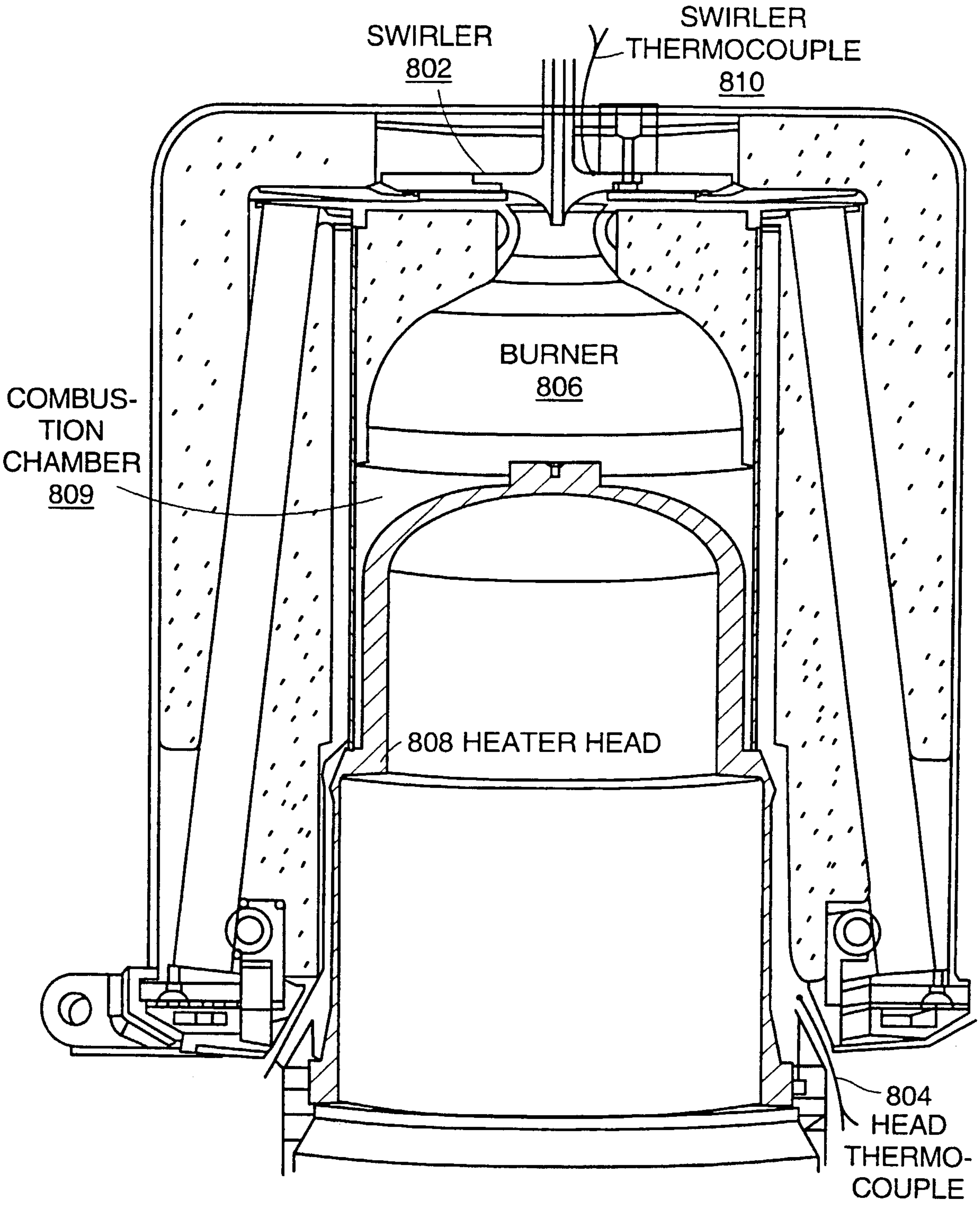

SWIRLER
802
SWIRLER
THERMOCOUPLE
810
BURNER
806
COMBUSTION
CHAMBER
809
808 HEATER HEAD
804
HEAD
THERMOCOUPLE

FIG. 8

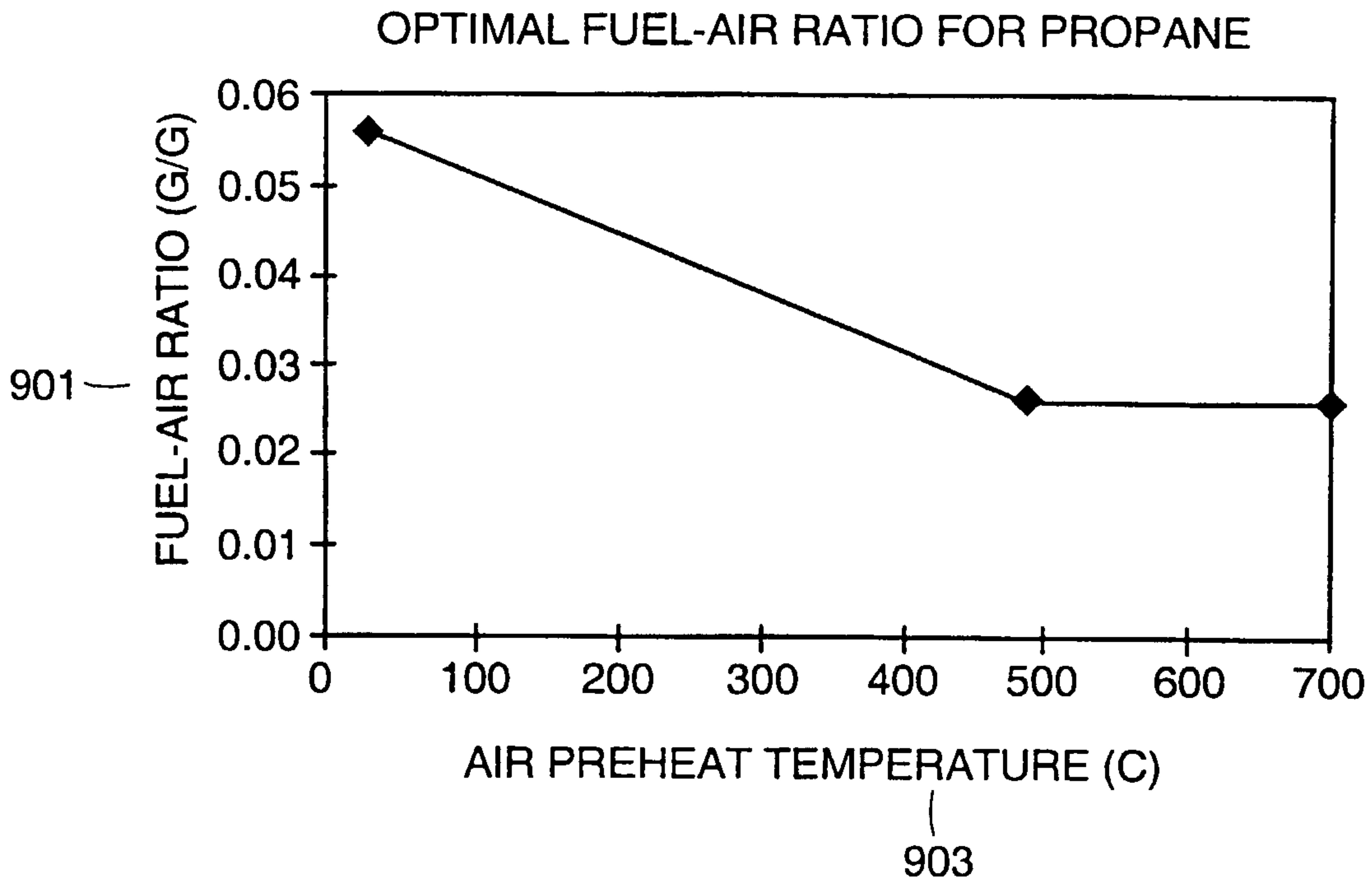

OPTIMAL FUEL-AIR RATIO FOR PROPANE
FUEL-AIR RATIO (G/G)
0.06
0.05
0.04
0.03
0.02
0.01
0.00
0
100
200
300
400
500
600
700
AIR PREHEAT TEMPERATURE (C)
901
903

FIG. 9


724 FUEL REGULATOR
699
706
703
704
714
7c
7c
7b
7b
701
702
707
722
708
716
711
712
AIR 700
AIR 700
BLOWER 728
709 FUEL & AIR
710
BURNER CONTROLLER 726
720