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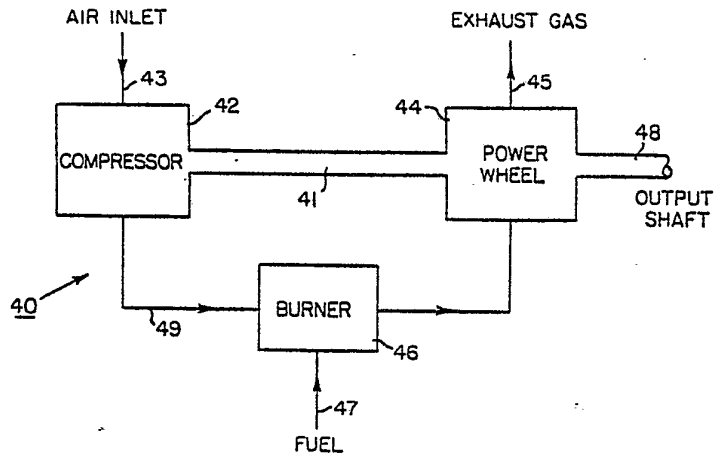
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(57) Abstract

A positive displacement, constant pressure Brayton cycle engine (40) including a compressor (42), a combustion chamber (46), a motor (44) and means for operating the engine in a substantially constant pressure Brayton cycle including means for continuously heating the gas in the combustion chamber (46) at constant pressure and means for feeding a continuous stream of hot, compressed gas at constant pressure solely to the motor (44). Preferably one or both of the compressor (42) and motor (44) are positive displacement rotary sliding vane devices. The engine (40) can be operated with an intermittent hot-cool operating cycle and with various throttle control systems.

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ENGINE SYSTEM

TECHNICAL FIELD

This invention relates to engines or prime movers and more particularly to a constant pressure, positive displacement, Brayton cycle engine.

BACKGROUND OF PRIOR ART

All previous methods of adapting the Brayton (or Joule or constant pressure) continuous cycle for prime mover service have centered around the classical gas turbine concept. A gas turbine includes three devices: A dynamic (non-positive displacement) compressor of the axial or centrifugal type; a fuel combustor of the direct or indirect type; an output turbine (power wheel) of the impulse or reaction type. The gas turbine approach has many advantages at high power levels (central station power generation, aircraft engines, etc.), but suffers serious economic handicaps at the lower horsepower levels (up to about 300 h.p.) required by automobile service.

The compressor and turbine share a common shaft in the classical gas turbine engine. Thus, the turbine (in a single shaft engine) must drive its own compressor in addition to driving the load. The reason for this is that only a gas turbine is capable of the ultra-high r.p.m required by the previously used, high speed, dynamic compressors. A simple, single shaft, classical engine is shown in Fig. 1 and comprises a compressor 12, a turbine 14, a shaft 16 connecting the compressor and turbine and a combustion chamber (or burner) 18. Figs. 2A and 2B show two cases of different compressor-to-output power ratios (Pc/Po). Fig. 2A shows a compressor 20 and a turbine 22 connected together by a shaft 24. The arrangement shown in Fig. 2A has a compressor-to-output power ratio of 0.5. Thus, a 100 h.p. output requires a 0.5 x 100 = 50 h.p. compressor. It follows that the power turbine for this case must supply 50 + 100 = 150 h.p. Fig. 2B shows a compressor 26 and a turbine 28

BUREAU OMPI WIPO INTERNATIONAL

connected together by a shaft 30. In Fig. 2B the compressor-to-output power ratio is 2.0. The 100 h.p. output power for this case requires a 200 h.p. compressor and a 100 + 200 = 300 h.p. turbine. Clearly the overall equipment size, for a given output shaft power, depends upon the required compressor size. It can be shown (see "Aircraft Gas Turbines" C. W. Smith John Wiley at page 43, or "Propulsion Systems" A. N. Hosny, University of South Carolina Press at page 51) that the compressor-to-output power ratio can be simply stated as:

$$1. \quad \frac{Pc}{Po} = \frac{1}{\eta_T \eta_C \frac{1}{x} \left(\frac{T_3}{T_1}\right) - 1}$$

where:

Pc/Po = Compressor To Output Horsepower Ratio

T_3/T_1 = Combustion To Inlet Temperature Ratio (°F, Absolute)

η_T = Thermodynamic Efficiency of Turbine

η_C = Thermodynamic Efficiency of Compressor

$x = \left(\frac{P_2}{P_1}\right)^{\frac{\sigma-1}{\sigma}}$

P_2/P_1 = Compressor Outlet To Inlet Pressure Ratio

σ = Ratio of Heat Capacity At Constant Pressure To That At Constant Volume

For a fixed ambient temperature (T_1), equation 1 states that the compressor-to-output power ratio depends only upon the pressure ratio adapted and the combustion temperature, since the other parameters, η_T, η_C, and σ are essentially fixed. Equation 1 is plotted in Fig. 3 and provides a good overall picture of the classical problem. Anything above a compressor-to-output power ratio of 1.0 means a very large compressor indeed. The parameters from Fig. 3 can be transformed to practical values in order to illustrate the classic problem involved and the solution provided by this invention. Consider the case of 50 h.p. net output.

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The data from Fig. 3 can now be translated to absolute compressor ratings and this is plotted in Fig. 4. Typical materials in today's technology allow for a temperature ratio (T_3/T_1) of about 3. Thus, Fig. 4 shows a marked sensitivity to pressure ratio. Absolute efficiency considerations dictate a pressure ratio in the range of 4:1 to 6:1. If a temperature ratio of 6 could be tolerated, Fig. 4 shows an insensitivity of compressor requirements to pressure ratio. If we assume an ambient temperature of 70°F, then a clearer picture of the problem emerges and is shown in Fig. 5. The steepness of the curve in the vicinity of 1000°F is the cause of the major problems as will become clear from the next curve.

A careful review of commercially available compressors of different types allows the compressor weight vs. horsepower curves of Fig. 6 to be drawn. The high r.p.m. associated with classical gas turbines limits severely the tolerable turbine inlet temperature. The relatively cheap materials that must be used for automobile service sets a limit of 1000-1500°F. Fig. 5 dictates a compressor in the 50-100 h.p. class to supply a net output of 50 h.p. Consider now Fig. 6: A piston compressor is completely out of the question since it would weigh between 1500 and 2000 pounds. The Roots type blower and sliding vane type compressors are better but also are in the impractical range of 200-600 pounds to supply 50 h.p. of shaft output power. Only the non-positive displacement compressor has a reasonable weight for the classical approach. It is this compressor weight factor alone - not cost nor efficiency but simply compressor weight that forces classical gas turbines to use the non-positive displacement compressor to supply the massive amounts of compressed air that are required.

The penalty paid for achieving the high compressor h.p. to weight ratio for the classic gas turbine is severe indeed:

1. The ultra-high r.p.m. is many times greater than the automobile r.p.m.

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2. The ultra-high r.p.m. of the hot turbine gives rise to excessive tensile stress due to centrifugal forces. This limits both usable materials and allowable inlet temperatures.

3. The compressor mass flow and output pressure vary with r.p.m. The fact that pressure varies means part-load efficiency is poor.

It is important to note that the output turbine requirement does not dictate a high r.p.m. per se; the high r.p.m. is required strictly to satisfy the compressor needs. In fact, there is no other method (electric motor drive, for example) of achieving the high r.p.m. required by the compressor.

This then is the state of affairs for gas turbines in automobile service. Fuel economy dictates both a high inlet temperature and a relatively high pressure ratio (about 4:1). Material limitations prevent high combustion temperatures, which in turn results in both low efficiency and high compressor requirements (Fig. 5), thus practically eliminating all but high speed, turbo-compressors (Fig. 6) on a weight basis alone. It is important to note that jet aircraft, central station power, and other non-auto applications can afford to use more expensive materials and devices and hence can utilize somewhat higher combustion temperatures. A jet aircraft turbine may cost $50,000 to build, but an automobile gas turbine must be limited to the $100-$200 class. As a result, gas turbines have found widespread service outside the automobile field only.

The main thrust of gas turbine technology today is towards better and cheaper high temperature materials and assemblies, super-alloys, exotic cooling methods, ceramic turbine buckets, cermet technology, plasma coatings, cheaper heat exchangers, etc.

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BRIEF SUMMARY OF THE INVENTION

The present invention is a constant pressure, positive displacement Brayton cycle engine method and apparatus. The Brayton cycle engine of the present invention includes a source of compressed gas (in a preferred embodiment the source is a positive displacement, rotary sliding vane compressor connected to the combustion chamber through a surge tank), a motor (in a preferred embodiment the motor is a positive displacement, rotary sliding vane motor), a combustion chamber, means for feeding compressed gas from the source to the combustion chamber, means for continuously burning fuel in the combustion chamber and continuously heating the gas, means for continuously feeding hot, compressed, constant pressure gas from the combustion chamber to the motor, means for feeding exhaust gas from the motor, and means for maintaining a constant gas pressure during the burn phase in the combustion chamber up to the motor inlet.

This removes the classic problems associated with high speed, non-positive displacement compressors: high cost and complexity, high stress on hot turbine buckets, low efficiency at part-load, etc. Use of a low speed, positive displacement compressor and motor is made possible by the use of high combustion temperatures as shown in Fig. 5. These high combustion temperatures are basically made possible by either one of two methods:

1. Use of an intermittent hot-cool operating cycle, either that described below with reference to Figs. 22-73, or to the inherent hot-cool operating cycle of the rotary sliding vane device; and

2. Use of a bootstrap argument. This results from the basic understanding provided by Figs. 1 through 6. The adoption of a lower r.p.m. means a lower tensile stress

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on the hot turbine bucket. The reduced stress permits an increased temperature since high temperature creep is a main limitation. The increased temperature results in reduced compressor requirements (Fig. 5) which in turn makes positive displacement compressors (low r.p.m.) feasible. The net result is a cheaper, more efficient and flexible implementation of the Brayton cycle.

Of course, as new high temperature materials, devices, and techniques became available they can be incorporated into the present inventions. However, it is to be clearly understood that the present invention can be practiced with today's materials, devices, and technologies.

The engine of the present invention can be operated in an open cycle arrangement in which the exhaust gas is simply fed to atmosphere directly or indirectly, or in a closed cycle arrangement in which the exhaust gas is fed back into the gas inlet port. In both embodiments the exhaust gas is preferably heat exchanged with the gas fed to the combustion chamber inlet.

It is an object of the present invention to overcome the disadvantages of the Brayton cycle gas turbine by providing a non-gas turbine implementation of the constant pressure Brayton cycle. It is another object of the invention to provide a constant pressure, positive displacement, Brayton cycle engine.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be more fully understood by reference to the following detailed description thereof, when read in conjunction with the attached drawings, wherein like reference numerals refer to like elements and wherein;

Fig. 1 is a partly diagramatic, partly schematic view of a prior art Brayton cycle gas turbine;

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Figs. 2A and 2B are diagramatic views of a compressor-turbine-shaft combination;

Fig. 3 is a graph of the ratio of compressor power to output shaft power vs. pressure ratio.

Fig. 4 is a graph of required compressor power (h.p.) vs. pressure ratio;

Fig. 5 is a graph of required compressor power (h.p.) vs. combustion temperature (°F);

Fig. 6 is a graph of compressor weight (lbs.) which is essentially compressor power (h.p.);

Fig. 7 is a partly diagramatic, partly schematic view of an engine according to one embodiment of the present invention including a compressor, a motor, and a combustion chamber;

Fig. 8 is a partly diagramatic, partly schematic, view of an engine according to another embodiment of the present invention including a compressor, a motor, and an indirect combustion chamber for use in a closed cycle operation;

Fig. 9 is a graph showing desirability vs. horsepower, speed, or torque;

Fig. 10 is a graph showing tensile strength vs. temperature for selected graphites;

Fig. 11 is a graph showing vaporization rate vs. temperature for graphite;

Fig. 12 is a graph showing pressure and flow rate vs. r.p.m. for a dynamic turbo-compressor;

Fig. 13 is a graph showing pressure and flow rate vs. r.p.m. for a positive displacement compressor;

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Figs. 14A, 15A and 16A are each a partly diagramatic, partly schematic, partly cross-sectional view of a compressor-motor engine according to three different embodiments, respectively, of the present invention;

Figs. 14B, 15B, and 16B are each partly diagramatic, partly schematic, partly cross-sectional view of a compressor-motor engine according to three different embodiments, respectively, of the present invention;

Fig. 17 is a partly diagramatic, partly schematic, partly cross-sectional view of a compressor-motor engine according to another embodiment of this invention;

Fig. 18 is a partly diagramatic, partly schematic view of a compressor-motor engine according to another embodiment of the present invention;

Fig. 19 is a partly diagramatic, partly schematic view of a compressor-motor engine according to still another embodiment of the present invention;

Fig. 20 is a partly diagramatic, partly schematic view of a compressor-motor engine according to a further embodiment of the present invention; and

Fig. 21 is a partly diagramatic, partly schematic view of a compressor-motor engine according to a still further embodiment of the present invention.

Figs. 22-35 are graphs and schematic diagrams that will be useful in an understanding of one aspect of the present invention when studied in conjuction with the detailed description below;

Figs. 36-48 are partly schematic, partly diagrammatic drawings of single shaft gas turbine embodiments according to certain aspects of the present invention;

Figs. 49-55 are partly schematic, partly diagrammatic drawings of free turbine embodiments according to certain aspects of the present invention;

Fig. 56 is a partly schematic, partly diagrammatic drawing of a single shaft, two turbine, combined cycle embodiment according to one aspect of this invention;

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Fig. 57 is a partly schematic, partly diagrammatic drawing of a single shaft, plural turbine, plural burn chamber embodiment according to one aspect of this invention;

Fig. 58 is a graph illustrating an operating cycle of the turbine of Fig. 57;

Fig. 59 is a partly schematic, partly diagrammatic drawing of a two shaft, free turbine with 6 turbines and plural burn chambers embodiment according to one aspect of this invention;

Fig. 60 is a plan view of a flywheel;

Figs. 61-71 are additional graphs that will be useful in further understanding the present invention when studied in conjunction with the detailed description below;

Fig. 72 is a partly schematic, partly diagrammatic drawing of a gas turbine of the present invention with a control system and electric traction motor drive; and

Fig. 73 is a partly schematic, partly diagrammatic drawing of a simple manual gas turbine according to another embodiment of this invention;

Fig. 74 is a partly schematic, partly diagrammatic drawing of a gas turbine according to one embodiment of the present invention;

Figs. 75A-75D are graphs showing various combustion processes according to the present invention;

Figs. 76A and 76B are graphs showing a combusion operation responsive to a gradual increase in throttle demand according to one embodiment of the present invention;

Figs. 77A and 77B are graphs showing a combustion operation responsive to a gradual increase in throttle demand according to another embodiment of the present invention;

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Figs. 78-82 are graphs showing additional combustion operations according to additional embodiments of the present invention, and

Fig. 83 is a graph showing a combustion operation according to a still further embodiment of the present invention.

DETAILED DESCRIPTION OF INVENTION

With reference to the drawings, Fig. 7 shows a substantially constant pressure Brayton cycle engine 40 according to the present invention comprising a positive displacement, constant outlet pressure compressor 42 and a positive displacement constant inlet pressure motor 44 connected to the compressor by a shaft 41 and also having an output shaft 48. The engine 40 included a combustion chamber 46 to which fuel is fed by a fuel line 47. Air is fed into the compressor 42 through an air inlet 43 and compressed air is fed from the compressor 42 to the combustion chamber 46 through an air line 49. Compressed, hot gas is fed from the combustion chamber 46 through a continuation of the gas line 49 to an inlet to the motor 44 from which the exhaust gas is fed from an outlet to ambient through an exhaust line 45. The engine 40 is an open cycle engine as will be clearly understood by those skilled in the art.

Fig. 8 shows a closed cycle engine 50 according to another embodiment of the present invention. The engine 50 includes a positive displacement, constant outl pressure compressor 52 and a positive displacement, constant inlet pressure

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motor 54 connected to the compressor by a shaft 56. The motor 54 is also connected to an output shaft 60. The engine 50 also includes an indirect combustion chamber 58 into which air and fuel are fed by lines 68 and 70 respectively, and from which the exhaust gas is fed to ambient through an exhaust line 72. Compressed gas from the compressor 54 is fed by a line 62 from the compressor to a heat exchanger 74 (such as a coil) in the indirect combustion chamber 58 and from there to the motor 54. The exhaust from the motor 54 is fed by a line 64 back to the compressor 52. The exhaust gas from the motor 54 is heat exchanged by a heat exchanger 66 with the gas fed from the compressor 52 to the combustion chamber 58. The various advantages of the closed cycle are described below. It is to be noted that in the open cycle embodiment of Fig. 7, the hot gas fed to the output power wheel is the products of combustion of the burning fuel in air. In the closed cycle embodiment of Fig. 8, on the other hand, the hot gas fed to the output power wheel is whatever is chosen for the working gas, such as nitrogen, neon, carbon dioxide, etc.

According to the present invention, the engines 40 in Fig. 7 and 50 in Fig. 8 employ positive displacement, rotary sliding vane devices as either or both of the compressor and the motor. A rotary sliding vane compressor can be used with a dynamic turbine or with a rotary sliding vane motor, and the rotary sliding vane motor can be with various compressors including preferably a rotary sliding vane compressor. The compressors can be used alone or in combination with a surge tank having means to provide a constant outlet pressure and check valves as necessary to prevent any gas flow from the combustion chamber back toward the source of the constant pressure compressed air.

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Thus, the various possible combinations are:

1. constant pressure source/rotary vane
2. rotary vane/rotary vane
3. rotary vane/dynamic turbine

Of course, the rotary vane compressor is just one, although the preferred one, of the possible constant output pressure sources, used either alone or in combination with a constant output pressure surge tank. In the above, the first name refers to the compressor and the second to the motor.

These configurations result in a desirability curve vs. horsepower and/or speed and/or torque as depicted in Fig. 9.

One preferred specific embodiment is to use a rotary sliding vane motor with the vanes being made of graphite. This offers several advantages. Graphite is a good lubricant in itself and thus the wear on the vanes is minimized. Second, graphite is known to increase in tensile strength as temperature is increased to about 4500°F

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as shown in Fig. 10. Thus, this feature allows combustion temperatures up to about 4500°F to be used in the expansion output wheel. Fig. 10 is a graph of ultimate tensile strength vs. temperature for selected graphites. All specimens were tested in the direction of major anisotropy. (1) Petroleum coke base, fine grain, extruded, d = 1.67; (2) lampblack base, molded, d = 1.50; (3) petroleum coke base, medium grain, extruded, d = 1.55; (4) petroleum coke base, fine grain, molded, d = 1.75.

Graphite is known to have a relatively high vapor pressure at high temperatures. This will result in a vaporization rate. Assuming 100,000 miles of service life, and 50 miles/hour average speed, then 7×10^6 sec. of service life is required. If 10% of the graphite surface is allowed to vaporize during this period, then a vaporization rate of about 10^{-8} gram/cm^2 sec. is tolerable. From Fig. 11 (which is the free vaporization rate of graphite) it is seen that an average temperature of about 3600°F (2200°K) is tolerable. While this temperature greatly exceeds present day gas turbine capabilities, it can be extended further by coating the graphite vanes with a suitable high temperature metal, alloy, or ceramic. In this case, vaporization will occur (if at all), only at the tiny exposed area where the vane rubs the housing.

While the above discussion has been with respect to using graphite for the sliding vanes in the motor of the present invention it is also desirable to use graphite for the sliding vanes in a sliding vane type compressor. Graphite has the advantages that it wears uniformly and conforms by the wear process to the desired shape and that it also provides self lubrication, is relatively inexpensive and is easy to machine, etc.

Another method of causing a major reduction in the compressor size and weight is to utilize the closed cycle in Fig. 8 This technique does not change the

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compressor horsepower requirement. This fact is evident from equation 1. The compressor-to-output power ratio depends only upon the pressure ratio and not the absolute gas pressure. However, at higher total pressures made possible by the closed gas system, the compressor "swept out volume" becomes less due to the higher gas density. As a result, the key item (the compressor weight) is reduced. This allows for an extension of this invention to higher power levels. Thus, for example, the vane/vane desirability curve can now take the dashed position shown in Fig. 9.

It should be noted that a turbine does not require the same high tip speed (high r.p.m.) as does the turbo-compressor. Thus, a combination such as a vane/turbine is a useful configuration. While the main thrust of this invention is the automobile, it is to be understood that the engine of this invention is also applicable to trucks, locomotives, central station power, remote station power generation, emergency and stand-by power generators, aircraft, etc.

The present invention can be used with multiple shaft arrangements, multiple stage compressors, multiple stage expansion stages (motor output), interstage cooling, re-heat, and various techniques of heat exchange.

One major advantage of the present invention over the classical gas turbine for automotive service is in the matching of r.p.m. The wheels of an automobile rotate at a maximum of about 1500 r.p.m. Thus, complicated speed reducers are required to match a gas turbine, at 50,000 r.p.m., to an automobile. The r.p.m. of the present invention can be selected to closely match that of the automobile by proper selection of compressor and power wheel diameters.

Another major advantage of this invention over the prior art is the efficiency at part-load. In fact, the present teachings maintain efficiency essentially down to 0 r.p.m. This statement can best be visualized with

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the aid of Figs. 12 and 13 The classical turbo-compressor relies upon high impeller tip speed to provide both mass flow and pressure. Thus, a reduction in r.p.m. causes both mass flow rate and pressure to be reduced as shown in Fig. 12 The flow rate reduction at low r.p.m. is acceptable since low r.p.m. (part-load) requires only a portion of the air mass flow. However, efficiency considerations insist upon a high gas pressure, in fact, optimal conditions require a narrow band of gas pressures (pressure ratio in 4-6 range). Thus, the low r.p.m. drops the pressure ratio and introduces the well-known part-load inefficiency. This is not true for the positive displacement compressor of the present invention as shown in Fig. 13. The compressor outlet pressure does not depend upon speed but only upon the geometry of the compressor apparatus. The flow rate is a direct linear function of the r.p.m. This is the ideal state of affairs for part-load. As more fuel is pumped to the combustor, the gas motor (sliding vane, turbine, etc.) provides more output power and higher r.p.m. The higher r.p.m. provides more compressed gas, as it should, at the same pressure and thus the efficiency, and combustion temperature conditions do not change.

The following chart may be found useful:

TYPICAL COMMERCIAL UNITS

Mfg.	Type	Horsepower		Weight (pounds)	Model No.
		Room Temp.	2500° F		
Gast	Rotary vane	4.0	20.0	21.0	6AM-FRV-23A
Gast	Rotary vane	10.0	50.0	80.0	16AM-FRV-13

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The preferred structure of the compressor and of the motor of the present invention will now be described in more detail, with reference to Figs. 14-17. Figs. 18-21 show various engine systems using any one of the compressor-motor structures of Figs. 14-17.

Fig. 14Ashows a single, unitary compressor-motor 80 comprising a stator 82 having a chamber 84, and a rotor 86 rotatably mounted in the chamber 84 on a shaft 88. The rotor 86 includes a plurality of sliding vanes 90, in this case, four. The rotor is tangent to the stator at two locations and the rotor and stator define therebetween two working chambers, in this embodiment, a compressor chamber 92 and a motor chamber 94. The compressor chamber has a continuously decreasing volume in the direction of rotation of the rotor, and the motor chamber has a continuously increasing volume in the direction of rotation of the rotor (see arrow 96). The stator includes a compressor inlet 98 and outlet 100, and a motor inlet 102 and outlet 104. The vanes 90 are preferably graphite and compression springs 106 can be used to bias the vanes outwardly.

Fig. 15A shows a compressor-motor 107 similar to that of Fig. 14A, the difference being that in Fig. 15A the motor chamber 108 is larger than the compressor chamber 109, for reasons that will be clear and have been discussed elsewhere.

Fig. 16A shows an engine 110 similar to those in Figs. 14A and 15A except that in this embodiment the rotor is tangent to the stator at only one location. The engine 110 includes a stator 112 and a rotor 114 mounted eccentrically in the stator 112 on a shaft 116. The rotor has a plurality (four in this particular embodiment) of sliding vanes 118, received in slots 120 in the rotor, and has a small compression spring 122 in each slot for biasing the vanes radially outwardly to permit permit operation at very low R.P.M.

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The stator 112 has a plurality (four in this embodiment) of gas ports in fluid communication with a chamber 24 between the rotor the inner wall 125 of the stator. The ports include a cool (ambient) air inlet port 126, a compressed air outlet port 128, a hot gas inlet port 130, and an exhaust gas port 132.

An air line 134 feeds the compressed air from the port 128 to a combustion chamber 136 where it is burned with fuel fed thereto via a fuel line 138. The gaseous products of combustion are fed by a line 140 from the combustion chamber 136 to the port 130.

The rotor 114 turns counterclockwise in Fig. 16A (in the direction of the arrow 142). In addition, Fig. 16A shows another aspect of the present invention in which fuel can be fed into the inlet port 126 along with air for purposes that will be discussed in more detail below.

Another aspect of the present invention is shown with references to Figs. 14B, 15B and 16B. It will be seen that these devices are identical to those of Figs. 14A, 15A, and 16A except that in the "B" cases, the gas in the device is not always in fluid communication with either the inlet or the outlet port. The "B" case devices have more vanes and thus the gas as it goes through the device is, for a short period of time, sealed off from the outside world. However, even in these "B" cases, these are Brayton cycle engines because there is constant pressure during the burn time in the combustion chamber and from the inlet to the combustion chamber, through the combustion chamber, and to the motor inlet.

Referring now to Figs. 14B, 15B, and 16B the same elements have the same numerals as in Figs. 14-16 except that in the "B" case, a "/" prime is added to the corresponding numeral from the "A" case.

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A still further embodiment of the present invention is shown in Fig. 17 in which all elements have the same numeral as in Fig. 16 B except that a " // " double prime is added. This embodiment differs from that of Fig. 16B in that: (1) the stator 112" is oval or oblong so that the total cavity volume on the right (motor) side is greater than the total cavity volume on the left (compressor) side, to solve the unequal volume problem, and (2) a short spacing 119 is provided between the outer edge of each of the vanes 118" and the inside surface of the stator 112". The purpose of this spacing is to provide a continuity of flow of the gases in passing through the engine 110", so that the gas is never cut off of from the outside world. This spacing 119 is greater than about .003 inch to allow passage of gases therethrough. Various ways of providing this spacing can be used such as to provide a recess in the central portion of the end of the vane so that the two outer ends only of the approaches contact with the stator.

The net result in the embodiment of Figs. 14-17 of the present invention is that the same single stator, rotor and shaft provides both the compression function and the expansion function required in a Brayton cycle (continuous, constant pressure). It is to be noted that in order for an engine to operate in a constant pressure Brayton cycle, there must be continuity of flow during the burn or combustion phase of the cycle, that is, from the inlet to the combustion chamber, through the combustion chamber, and to the motor inlet, to provide a constant pressure during this period of time. That is, the gas must never be trapped but must be able to expand at constant pressure. The term "substantially constant pressure Brayton cycle" as used in the present application means such a cycle in which the gas pressure is substantially constant in the phase

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of the cycle between points 2 and 3 in Fig. 64A, that is, during the burn period the gas is never trapped but is always able to continuously expand at constant pressure. This phrase includes a changed pressure in that if the inlet pressure to the combustion chamber changes, then this changed, higher or lower, pressure is also maintained throughout the burning phase at such new higher or lower pressure between points 2 and 3 in Fig. 64A. Preferably, the source of compressed air and the motor have constant outlet and inlet pressures, respectively, such as when using rotary sliding vane devices therefor. When using a surge tank, the regenerator (or heat exchanger from the motor exhaust) can be built into the surge tank, so that the outlet pressure of the pressurized gas to the combustion chamber can vary with the amount of heat exchanged. Further, in Figs. 14A, 15A and 16A the gas in the engine is always either in communication with either the inlet or the outlet duct because the number of vanes is such that there is a maximum of one vane between the inlet and the outlet port, and for all orientations of the rotor. In the embodiment of Figs. 14B, 15B and 16B, two or more vanes are located between the inlet and outlet, and gas can be trapped therebetween and sealed off from the outside world. In Fig. 17, even though more than one vane can exist between the inlet and outlet duct, the opening 119 allows for communication of the gas with the outside world (meaning with the gas lines or ducts connected to the stator) at all times.

The configurations of Figs. 14-17 have several objects. First, maximum efficiency in the motor portion

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will occur if all the gas pressure is expanded against the vanes. This requires the expanded gas pressure to be just ambient pressure, with none left over, just as the gas reaches the exit port 32. Since the hot gas is several times less dense than the cool gas, the total cavity volume on the motor side should be greater than the cavity volume on the compressor side. This can be done by making the outer housing oblong as shown in Figs. 15A, 15B, and 17.

Operation at full efficiency is automatic at reduced load as can be visualized from Fig. 13. The classical gas turbine suffers a major efficiency loss at part-load since the compressor outlet pressure drops at reduced R.P.M. This is not true for a positive displacement compressor since gas pressure does not depend upon speed. In Fig. 13 it is seen that gas flow is a direct function of speed as is required in practice. Thus, a low throttle (low fuel burn) demands a low gas flow rate and a R.P.M. As a result, combustion temperature (efficiency) is invariant with throttle setting and R.P.M.

For an open cycle machine, the fuel may be injected into the gas inlet 126 (see Fig. 16A). Several purposes are served: The oil is vaporized and hence helps cool the vanes, the oil (liquid or vapor) will tend to lubricate the vane where and if it rubs on the housing, and an excellent air/fuel mixing action takes place that assures efficient combustion. Other liquids such as water, fuels, etc. can be injected into any or all of the chambers through ports 126 or 130 or through any other special inlet orifices.

Each half revolution, a vane is heated, and each half revolution each vane is cooled. Thus, a built-in burn/cool (or hot/cool) cycle is provided without interrupting the actual burn. These periods are such, at low and high R.P.M.'s, to assure no high peak or average temperatures will occur in either the vanes or the structure (other than that immediately adjacent the hot gas inlet to the motor).

BUREAU OMPI WIPO INTERNATIONAL

The unitary devices of Fig. 14-17 of the present invention also serves as a heat exchanger since the hot exhaust gases are in thermal contact with the inlet gases via the structure. Structure design is such as to maximize this effect. It is noted that the natural geometry is right, that is, the hottest section of the motor section is adjacent to the hot compressor outlet side.

In open cycle operation using the engine of Fig. 16A, care must be exercised to assure complete evacuation of the exhaust gas from the cavity. It may be desirable in Fig. 16A to connect a low pressure fan or blow at the inlet (port 126) to ensure that all of the exhaust combustion gases (starved in oxygen) are evacuated.

The evacuation problem of Fig. 16A does not exist for the closed cycle case (described below with reference to Fig. 21) since the exhaust and inlet gases are one and the same. This is one advantage of using a closed cycle.

Fig. 18 shows an engine system 180 according to another embodiment of the present invention including a single, unitary, positive displacement compressor-motor 182 (the circle 182 is intended in Figs. 18-21 to represent a rotary sliding vane compressor-motor according to the present invention, such as shown in any of the different embodiments of Figs. 14-17). The compressor-motor 182 has an air inlet 184, a compressed air outlet 186, a line 188 for feeding compressed air to a compressed air or surge tank 190, a line 192 for feeding compressed air from the surge tank 190 to a regulating valve 194 from which the compressed air is fed to a combustion chamber 196, where it is burned with fuel fed to the combustion chamber 196 through a fuel line 198 and a regulating valve 200. The hot combustion gases are fed solely from the combustion chamber 196 through a gas line 202 to a hot gas inlet 204 of the compressor-motor 182. The exhaust gas is fed from an exhaust port 206 and exhaust line 208 to ambient. The term "solely" means that gas from the combustion chamber cannot flow backward toward the source of compressed air. The valve 194 can

BUREAU OMPI WIPO INTERNATIONAL

include a check valve or one way valve. A throttle control 210 is connected to each of the regulating valves in response to demand, for example, as described below in the section on throttle control.

The surge tank is always charged up and can supply a sudden surge for acceleration and/or start-up purposes, in addition to supplying the constant pressure gas required for the Brayton cycle.

Special throttle systems can be used to handle the start from zero speed case. If a clutch or transmission device is used, then a small built in "slip" can permit a slight (perhaps 1 R.P.M.) motor rotational speed at zero vehicle velocity. If the motor is directly coupled to the vehicle, then a small gas flow (leakage through compressor/motor vane) is possible at zero vehicle speed thus assuming continuity of gas flow. Another possibility is a valve arrangement on the gas line shown in Fig. 18 that is essentially wide open at vehicle speeds above, say 1 m.p.h. and the throttle control becomes a function of fuel rate only. The slow acceleration, or speed of response, inherent in the classical gas turbine is not present in this invention. In the gas turbine case the lag is due to the moment of inertia of the turbine compressor assembly. This entire assembly must get up to, say, 50,000 R.P.M. before power can be developed. This is not the case in this positive displacement, direct drive engine.

A combination of low R.P.M. (low stress on blades) and the built-in hot/cool cycle (alone or in combination with the intermittent burn operation described below) will allow turbine inlet temperatures of 3500°F to be possible while average temperatures of the blades and structure will stay below 1000°F (as will the peak temperature). As a result, materials such as stainless steel, and super alloys will be useful. One preferred material for the blades is graphite, however, for a number of reasons.

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It is noted that commercial rotary air motors are built to be intentionally inefficient so as to not freeze-up water vapor in the air lines upon expansion. High efficiency requires a gradual expansion of the gas. However, in the prior art air motor all of the pressure drop is abrupt across the constant volume section. As a result, the entire pressure drop is across the outlet orifice and the motor is nothing but an inefficient "paddle-wheel". This is in direct opposition to the present invention.

The gases going through the motor portion of this invention are much less dense than those going through the compressor portion. As a result, more physical volume is required on the motor side in order that the hot gases can fully expand before leaving the exhaust port. One method of accomplishing this is to make the stator or housing cam-shaped.

Fig. 19 shows an embodiment of the present invention that can be used to balance the volume/pressure relationships between the compressor and motor sides of the compressor-motor. Fig. 19 shows an engine system 210 including a compressor-motor 212 similar to that shown in any one of Figs. 14-16 and having an air inlet port 218, a compressed air outlet port 220, a hot gas inlet port 222 and an exhaust gas outlet port 224.

In addition to this basic structure, the embodiment shown in Fig. 19 also includes a compressed air or surge tank 226 connected to the compressed gas feed line 228 from the compressed gas outlet 220 to the combustion chamber 230 by means of a surge tank line 232. An outlet conduit 234 is connected between the surge tank 226 and the compressor-motor 212 engine adjacent the hot gas inlet 222 at a supplemental inlet port 236. The compressed gas line 228 includes a regulating valve 238 and and the surge tank outlet line 234 includes a regulating valve 240. Fuel is fed to the combustion chamber 230 by a fuel line 242 controlled by a regulating valve 244.

BUREAU OMPI WIPO INTERNATIONAL

A symmetric relationship again is assumed in Fig. 19 with equal volumes for the compressor and motor sides. However, the total flow through the motor side is reduced by routing a portion of the compressor output to the storage or surge tank. At some later time the surge tank will be full, and at this time the fuel flow can be stopped, and the compressor-motor can be driven by the stored gas in the surge tank. Thus, fuel is saved, the compression energy in the stored gas is recovered, and in effect the physical capacity in the motor side has been increased relative to the compressor side.

Fig. 20 shows another embodiment of the present invention providing an alternative way of increasing the equivalent physical capacity of the motor side relative to the compresssor side of the compressor-motor. Fig. 20 shows an engine system 250 using a compressor-motor 182 similar to that shown in Fig. 18 with the exception that the engine 250 also includes a gas conduit 252 for routing a part of the compressor output gas back to the compressor inlet.

The embodiment of Fig. 20 increases the equivalent physical capacity of the motor side relative to the compressor side. In this case, a physical symmetry is again assumed. The routing of a part of the compressor output back to the compressor inlet, has the effect of increasing the compressor outlet pressure (pressure ratio) at the expense of total flow. Thus, in effect, the compressor physical volume is decreased since part of the gas is recirculated and passes through the compressor several times. As a result, the flow through the motor side has been increased. This process is well known in the compressor art and is often known as regeneration. Optimal operating conditions are set by an orifice pair arrangement that determines the compressor outlet split ratio between the combustor inlet and the compressor inlet. Proper precautions must be taken to prevent surge, starvation, and the like as

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will be understood by those skilled in this art. One advantage of the "regeneration" method is the flexibility of the point of injection. If near the motor exhaust valve, then the positive pressure can be used to help flush-out the exhaust gases from the motor section in the embodiment of Fig. 14. In addition, a portion of the gas routed back can be introduced at one point while other portions can be introduced at other different locations.

Fig. 21 shows a Brayton cycle engine system 260 according to the present invention including a positive displacement compressor and a single, unitary compressor-motor 182 similar to that shown in Fig. 18. The engine in Fig. 21 is operated in a closed cycle system by means of gas line 262 which feeds the exhaust gas from port 206 back to the inlet port 184 through a heat exchanger 264 which heat exchanges with the compressed gas flowing from the port 186 to the combustion chamber. In this embodiment, the combustion chamber is an indirect combustion chamber 266 and the compressed gas line from port 186 to port 204 includes a heat exchanger 268 in the form of, for example, a coil. Fuel and air are fed into the indirect combustion chamber 266 and the combustion products are exhausted therefrom as shown in Fig. 21.

Fig. 21 also shows another aspect of the present invention of a plurality of heat radiating fins 270 located at least on the motor side of the compressor-motor 182. These fins may, of course, be located over the entire surface thereof if desired.

Rotation in the drawings is counterclockwise with the vanes on the left side compressing the inlet gas from the compressor inlet port (such as 184) and exhausting compressed gas from the device at the compressor outlet port (such as 186). In an open cycle machine, this gas will be air. In a closed cycle machine, this gas will be whatever the working gas happens to be - nitrogen, neon,

BUREAU OMPI WIPO INTERNATIONAL

carbon dioxide, etc. The hot gas from the combustion chamber enters through the motor inlet port (such as 204), expands against the vanes, and exhausts through the motor exhaust port (such as 206). In an open cycle machine this hot gas is the product of combustion, e.g. products of burning oil in air. In a closed cycle machine these hot gases are whatever the working gas happens to be as discussed above.

Another aspect of the present invention is the combination of the constant pressure, positive displacment, Brayton cycle engine with the intermittent burn-cool operating cycle described below. The following description is also found in U.S. application Serial No. 879,969 filed February 21, 1978 entitled "Gas Turbine System" by the same inventor as is the present invention. While this description is primarily with reference to a Brayton cycle gas turbine, all embodiments thereof are clearly also applicable to the present invention of a positive displacement Brayton cycle engine, such as a vane/turbine or even a vane/vane arrangement. While the intermittent burn-cool operating cycle described below is not as important in the embodiments that use a rotary sliding vane motor as it is in the embodiments that use a turbine motor, it can still be used with useful results even with the rotary sliding vane motor. The reason it is not as important for the rotary sliding vane motor is that the rotary sliding vane motor has an inherent, built-in intermittent hot-cool operating cycle, as mentioned above. In the gas turbine, all of the blades are exposed to the hot gas all of the time in the normal operating method and all of the time during the hot phase of the intermittent hot-cool operating cycle described herein. However, this is not the case with a rotary sliding vane motor. It is only the vane located at the hot inlet that is exposed to the hot gas at any one time. This vane then cools during the remainder of one rotation until it returns to the location of the hot inlet port. This provides an automatic hot-cool operating cycle every revolution of the rotor for each vane. While this would not



provide enough useful cool time in a turbine because of its much higher R.P.M., it does provide substantial cooling time in a slower R.:.M. rotary sliding vane device.

SUMMARY OF THE INTERMITTENT HOT-COOL OPERATING CYCLE

A gas turbine method and apparatus is described using an intermittent burn operating cycle comprising a burn phase followed by a cool phase. This cycle includes: (1) feeding fuel to the burn chamber and burning it during the burn phase of the cycle, whereby the temperature of the chamber and turbine rises during the burn phase, and (2) cutting off most or all of the flow of fuel to the burn chamber during the cool phase whereby the temperature of the chamber and turbine falls. The air to fuel ratio during the burn phase is always the same providing a high combustion temperature and high efficiency. The part-load efficiency loss problem of the prior art is eliminated by the present invention. In the prior art, under part-load, the air-to-fuel ratio was increased, thus resulting in a reduced burning temperature and reduced efficiency. The combustion temperature is much higher than the temperature that the structure is allowed to reach because of the intermittent burn cycle and the cool down phase, thus eliminating the need for high temperature materials. In addition, lower amounts of air are needed as compared to the prior art gas turbine. For example, the gas turbine of this invention uses about six times smaller volume of gas than does the well-known Chrysler automobile gas turbine.

The gas turbine of the present invention also includes means for preventing the shaft speed from decreasing during the cool phase. This is accomplished by a flywheel action using, for example, a mechanical flywheel, an electric flywheel or a chemical flywheel. The present invention also includes the controlling of

BUREAU OMPI WIPO INTERNATIONAL

the length of at least one of the burn and cool phases to control the power output of the turbine. For example, if it is desired to increase power, the length of the burn phase can be increased, or the length of the cool phase can be decreased or both.

The present invention also involves the use of different amounts and pressures of air during the cool phase than are used during the burn phase. Another aspect of the present invention is the use of water injection during the cool phase to improve cooling. The water can be recovered in a condensor and recycled.

Another aspect of the invention includes the use of two different fuels, one, for example, can be used in the burn phase and the other in the cool phase. In another embodiment of the invention the shaft is connected to an electric generator and to a traction motor. A gear box, electric generator and battery can also be used in various combinations along with water injection. In addition, a heat exchanger can be used to pre-heat the compressed gas prior to it being fed into the combustion chamber.

The present invention also includes embodiments employing a free turbine. Combined cycle embodiments of the present invention include the use of a second working fluid operating with either an open or closed cycle and with a second working fluid operating to drive a turbine.

The present invention also includes embodiments using plural turbines and plural burn chambers in which the intermittent cycle is employed in each burn chamber. There is a staggered burn cycle from one burn chamber to the next which, in the preferred embodiment, is a uniform staggered burn cycle. A preferred air-to-fuel ratio is in the range of 14.7:1 to 35:1. A preferred burn phase to cool phase is 90% to 10% in length of time of the respective phases.

BUREAU OMPI WIPO INTERNATION

DETAILED DESCRIPTION OF THE INTERMITTENT HOT-COOL OPERATING CYCLE

The gas turbine is a heat engine and as such requires high combustion temperature and low exhaust temperature for good efficiency. An intensive, industry-wide effort to improve gas turbine efficiency is evident from the literature and the main attack is two-fold:

1. Better, cheaper heat exchange devices that thermally couple the hot exhaust gas stream to the cooler inlet gas stream are being designed. The desired effect is to reduce the heat energy loss via the exhaust stream.
2. New, high temperature materials are being sought for the critical structure members: combustion chamber, turbine blade, etc.

The Carnot principle states that the maximum available heat engine efficiency depends only upon the ratio of the combustion temperature to that of the exhaust medium:

1. $\eta_{max} = 100 \left(1 - \frac{T_e}{T_a}\right)$

η_{max} = Maximum Available Efficiency

T_e = Absolute Temperature of the Exhaust Medium

T_c = Absolute Combustion Temperature

The operation of a simple gas turbine is illustrated in Fig. 22. Ambient air is compressed, mixed with a fuel, burned, and allowed to expand causing a turbine wheel to rotate. For a fixed throttle setting the combustion temperature is constant and the Carnot efficiency is easy to estimate. A high throttle setting demands a high burn rate and hence a high combustion temperature. A low throttle setting demands a low burn rate and hence a low combustion temperature. Thus, the common comment of poor gas turbine efficiency at reduced load (low throttle setting). The lack of suitable high

BUREAU OMPI WIPO

temperature materials limits turbine efficiency at high throttle settings. Excess air is required at high load in order to reduce combustion temperatures. This hurts efficiency in two ways: first, excessive amounts of air must be compressed; second, the combustion temperatures are low thus hurting the Carnot efficiency.

One main feature of the present invention is to help both the low and high throttle efficiency problems by using an intermittent operation in which fuel is burned in a normal fashion for a period of time, for example, two minutes. Next, the fuel is cut off for a period of time, for example, 20 seconds. Cool, compressed ambient air continues to flow through the gas turbine during the "cool" period thereby reducing the temperature of the critical structural members. A typical set of graphs is shown in Fig. 23. It is clear that the average temperature of the critical structure members is less than the peak combustion temperature. Since the Carnot efficiency is determined by the peak combustion temperature, this is a method to legitimately "beat Carnot". That is, a high combustion temperature is achieved for efficiency purposes while exposing the critical materials to a much lower average temperature. Fig. 23 is highly idealized and ignores the rounding of corners that will occur due to thermal lag and the like. An important aspect of the present invention concerns these corner rounding effects, and this will be discussed below.

The top graph in Fig. 23 shows the fuel burn history: burn, cool, burn, cool, etc. The second and third graphs show the working gas and structural member temperatures, respectively. A constant or equilibrium temperature is reached during the burn phase and a lower equilibrium temperature is reached during the cool phase. Of course, different parts of the structure will reach different equilibrium temperatures, for example, the combustion chamber will run hotter than the turbine wheel.

BUREAU OMPI WIPO

A smooth and steady output is achieved according to one embodiment of the present invention by the use of a flywheel; this is discussed in detail below. A steady rotational speed (at a fixed throttle setting) is desired and is achieved by the present invention. The last two graphs in Fig. 23 reflect this fact in showing the rotational energy stored in the turbine wheel and power delivered to the load. The constant load output is used to overcome friction, grade, and aerodynamic drag and to accelerate the automobile. The fact that no fuel is spect during the "cool" phase means that the Carnot efficiency is determined only by the high peak combustion temperature.

While the top graph in Fig. 23 shows zero fuel burn rate during the cool phase, this is not essential. In fact, in the preferred embodiment there is a very small "keep-alive" fuel burn rate (as shown by the dotted line in Fig. 23A). Maximum burn time would be about 5-8 minutes.

Fig.24 shows the actual "rounded corner" situation. Fuel injection and fuel cut-off will have a small but finite time delay as shown in the top graph in Fig. 24. Several factors cause the gas temperature curve (the second graph in Fig. 24 to be rounded. Gas takes a small but finite amount of time to heat up; combustion is not perfectly uniform; the early heat must be shared with the structure, for example. The structural member temperature rounding (in the lowest graph in Fig.24) is more dramatic due to the higher thermal mass. It takes longer to heat a cubic foot of steel than it does a cubic foot of air.

The Carnot efficiency is not so easy to calculate for Fig. 24. Some energy spent during the burn phase is released during the cooling period and vice-versa. However, the following initial observations can be made that will prove useful in the detailed quantitative analysis that will follow:

BUREAU OMPI WIPO INTERNATIONAL

1. In one embodiment, a flywheel action is incorporated into the intermittent operation concept. This flywheel can be mechanical, electrical, or chemical.
2. The main characteristic demanded of the turbine materials used in the present invention is not high temperature resistance, but rather tolerance to cyclic temperature changes. Assume an engine lifetime of 100,000 miles at an average speed of 25 miles per hour. This requires 4000 hours of operation and amounts to 120,000 temperature cycles at 2 minutes per burn period.
3. The intermittent operation feature of this invention improves efficiency at all throttle settings high and low. The classical poor efficiency at reduced or part-load phenomena no longer exists.
4. A variable length of "burn" and/or of "cool" time between burns is the preferred form of throttle control.
5. The effects of rounded temperature corners upon efficiency will be beneficial in some cases and not in other cases.
6. All factors leading to rounded corners are important; the thermal mass of the turbine wheel is a good example. A high thermal mass soaks up much heat and takes a long time in doing so.

The following preliminary analysis will be helpful in understanding the present invention. A very simple, but accurate model will be used and the following assumptions apply:

1. Air is compressed and made to pass at a constant mass flow rate through the combustion chamber.

BUREAU OMPI WIPO INTERNATIONAL

2. Fuel is burned at a fixed mass flow rate.
3. The gas temperature rise is nearly instantaneous in the combustion chamber.
4. The walls of the combustion chamber, the turbine wheel, and the interconnecting ducts will rise in temperature and the hot gas will drop in temperature. A state of thermal equilibrium will be approached.
5. Things are reversed during the cooling phase. The gas temperature drops instantaneously in the combustion chamber to that of the compressor output. Now, the structure cools off and the gas heats up.

The effect of mass upon temperature rise for equal heating rates is shown in Fig. 25. The same heat rate is applied to the heavy mass (B) and the light mass (A) at the time "to". The light mass reaches equilibrium much faster and its equilibrium temperature will in general be less due to the higher surface area-to-volume ration. The dotted line in Fig. 24 shows the small "keep-alive" fuel burn rate, which is used in the preferred embodiment as discussed above regarding Fig. 23.

The effect of heating rate upon the temperature history of a fixed mass is shown in Fig. 26. Curve A is for a high input heat rate and B a low rate. The high rate curve not only responds faster but approaches a higher equilibrium temperature.

The following basic algebraic equations will prove helpful in understanding the later analysis.

BUREAU OMPI WIPO INTERNATIONAL

2. $\Delta Q = C\ W\ \Delta T$

ΔQ = Required Heat (Btu)

W = Mass (pounds)

C = Specific Heat of Mass (Btu/Pound-°F)

ΔT = Temperature Rise (°F)

Equation 2 applies to any body as long as melting does not occur and that a significant amount of heat is not given up by radiation, convection, etc.

Equation 2 can be differentiated to give the initial rate of change of temperature rise:

3. $\left(\frac{dT}{dt}\right)_o = \frac{\dot{Q}}{CW}$

$\dot{Q}$ = Input Heat Rate (Btu/minute)

$\left(\frac{dT}{dt}\right)_o$ = Initial Temperature Rise $\left(\frac{°F}{\text{minute}}\right)$

For a given heat input rate ($\dot{Q}$), simple exponential temperature rise assumed as shown in Fig. 27, time constant, τ, is defined as the time required to reach the final temperature if the initial rate did not decrease. We may write, for Fig. 27:

4. $(T - T_o) = K\ (1 - e^{-t/\tau})$

T = Instantaneous Temperature

T_o = Initial Temperature

T_f = Final Temperature

K = Constant

τ = Time Constant

The condition that the final temperature be T_f at infinite time requires

$T_f - T_o = K$, thus;

BUREAU OMPI WIPO INTERNATIONAL

$$5.\quad (T - T_o) = (T_f - T_o)\ (1 - e^{-t/\tau})$$

Differentiation of equation 5 gives:

$$6.\quad \frac{dT}{dt} = (T_f - T_o)\left(\frac{1}{\tau}\right) e^{-t/\tau}$$

and

$$7.\quad \left(\frac{dT}{dt}\right)_o = \frac{(T_f - T_o)}{\tau}$$

Equating equations 7 and 3 gives:

$$8.\quad \tau = \frac{CW}{\dot{Q}}\ (T_f - T_o)$$

Equation 5 can be written in a more convenient form:

$$9.\quad T = T_o + (T_f - T_o)\ (1 - e^{-t/\tau})$$

The case of heating the gas stream in the turbine is different. Heat transfer to the gas stream is rapid and a given heat input rate ($\dot{Q}$) will impart a fixed increase in temperature. Thus, equation 2 becomes:

$$10.\quad \Delta T = \frac{\dot{Q}}{C\dot{W}}$$

ΔT = Temperature Rise of Gas Stream
$\dot{Q}$ = Input Heat Rate (i.e., Btu/min.)
C = Specific Heat of Air
$\dot{W}$ = Mass Flow Rate of Air (Pounds/min.)

BUREAU OMPI WIPO

The above simple approximations will be used in the analysis to follow. A summary is given first.

SUMMARY OF DESIGN RELATIONS

The heat content of most liquid fuels are all about the same. A heat rate, $\dot{Q}$, corresponding to a typical gasoline will therefore by used throughout. The specific heat of most solid materials are approximately the same and an average value will be used. The same holds for the specific heat of gases. A summary of the relations to be used is given below. Some relationships are given in several forms to avoid confusion.

10. $$Q_g \left(\frac{\text{Btu}}{\text{gallon}} \right) = 127{,}600$$

11. $$Q_g \left(\frac{\text{Btu}}{\text{pound}} \right) = 20{,}750$$

12. $$\dot{Q}_g \left(\frac{\text{Btu}}{\text{hour}} \right) = 127{,}600 \, \dot{V} \left(\frac{\text{gallons}}{\text{hour}} \right)$$

13. $$\dot{Q}_g \left(\frac{\text{Btu}}{\text{minute}} \right) = 2127 \, \dot{V} \left(\frac{\text{gallons}}{\text{hour}} \right)$$

14. $$\dot{Q}_g \left(\frac{\text{Btu}}{\text{minute}} \right) = 20{,}750 \, \dot{W}_f \left(\frac{\text{pounds}}{\text{minute}} \right)$$

15. $$\dot{V} \left(\frac{\text{gallons}}{\text{hour}} \right) = \frac{S \text{ (miles/hour)}}{E \text{ (miles/gallon)}}$$

16. $$\dot{Q}_g \left(\frac{\text{Btu}}{\text{minute}} \right) = \frac{2127 \, S \text{ (miles/hour)}}{E \text{ (miles/gallon)}}$$

BUREAU OMPI WIPO INTERNATIONAL

17. $C\left(\frac{\text{Btu}}{\text{Pound-°F}}\right) = 0.15$ (metal, ceramic)

18. $C\left(\frac{\text{Btu}}{\text{Pound-°F}}\right) = 0.24$ (air)

19. $\Delta Q = C\,W\,\Delta T$

20. $T = T_o + (T_f - T_o)\ (1 - e^{-t/\tau})$

21. $\frac{dT}{dt} = \left(\frac{T_f - T_o}{\tau}\right) e^{-t/\tau}$

22. $\left(\frac{dT}{dt}\right)_o = \left(\frac{T_f - T_o}{\tau}\right)$

23. $\tau = \frac{CW}{\dot{Q}}\ (T_f - T_o)$

24. $\Delta T = \frac{\dot{Q}}{C\dot{W}_a}$

25. $\Delta T = 20{,}750\ \frac{\dot{V}}{CE}$

T = Gas or Structure Temperature (°F)
T_o, T_f = Initial and Final Temperature (°F)
τ = Thermal Time Constant (minutes)
$\dot{Q}$ = Input Heat Rate (Btu/minute)
C = Specific Heat (Btu per Pound-°F)
$\dot{V}$ = Gasoline Rate (Gallons per hour)
S = Vehicle Speed (Miles per hour)
E = Vehicle Fuel Consumption (miles/gallon)
W = Mass of Thermal Body (Pounds)
$\dot{W}_a$ = Mass of Air Flow Rate (Pounds/minute)

BUREAU OMPI

$\dot{W}_f$ = Mass of Fuel Flow Rate (Pounds/minute)

ϵ = Air to Fuel Mass Ratio (dimensionless)

γ = Fraction of Fuel Energy Used to Heat Gas Stream

ΔT = Temperature Change of Gas Stream

Q_g = Heat Content of Gasoline

$\dot{Q}_g$ = Rate of Heat Release From Gasoline

$\frac{dT}{dt}$ = Temperature Rate (°F per minute)

SYSTEM ANALYSIS

A well designed gas turbine, even by today's standards, has a relatively small temperature drop from the combustion chamber to the turbine wheel. Therefore, the entire thermal structure, combustion chambers, ducts, and turbine wheel assembly are lumped into one equivalent thermal mass for our analysis.

We need to estimate how much thermal energy the structure drains from the hot gas. To do this we need to estimate the hot gas temperature. Combining equations 18 and 25 gives equation 26 which is

$$26. \quad (\Delta T)_{gas} = 86{,}700 \frac{\gamma}{\epsilon}$$

plotted in Fig. 28. Air to fuel ratios from 50:1 to 100:1 and gas heat fractions (γ) from 0.8 to 1.0 are common today and result in a gas temperature rise in the range of 800 to 1700°F by Fig.28. Fig.28 illustrates another advantage of intermittent operation--the higher allowable gas temperatures are achieved with less air

BUREAU OMPI WIPO INTERNATIONAL

flow (lower air/fuel ratio). This helps both the practical and efficiency problems associated with high exhaust volumes.

The thermal requirements for the structural mass are calculated below. Equations 17 and 19 give:

27. $(\Delta Q)_s = 0.15\ W_s\ (\Delta T)_s$

W_s = Mass of Structure (Pounds)

$(\Delta Q)_s$ = Heat Required (Btu)

$(\Delta T)_s$ = Temperature Rise of Structure (°F)

Differentiation of equation 27 gives:

28. $\frac{dTs}{dt} = \frac{6.7}{W_s}\ \dot{Q}_s$

The burn cycle must be short enough that the structure temperature never reaches the high gas temperature during the burn phase. This requires the structural temperature versus time curve to be in the linear region always. A good assumption is the heat flow to the structure is proportional to the total heat flow under these conditions. That is,

29. $\dot{Q}_s = B\dot{Q}_g$

B = Function of Gas Flow to Structure Geometry, Operating Temperatures, etc.

The proper specification of B represents one of the important design results. If the total heat load consists of the gas and the structure, then:

BUREAU OMPI WIPO INTERNATIONAL

30. $\sigma + B = 1$, and

31. $\dot{Q}_s = (1 - \sigma)\,\dot{Q}_g$

Insertion of equation 31 into 28 gives:

32. $\frac{dTs}{dt} = \frac{6.7}{Ws}(1 - \sigma)\,\dot{Q}_g$

Equations 32 and 16 give:

33. $(\Delta T_s)_{Burn} = 14{,}251\,\frac{S\,(1 - \sigma)}{E\;W_s}\,(\Delta t)_{Burn}$

$(\Delta T_s)_{Burn}$ = Structure Temperature Rise During Burn (°F)

$(\Delta t)_{Burn}$ = Burn Time (Minutes)

W_s = Mass of Structure (Pounds)

S = Vehicle Speed (miles per hour)

ϵ = Fuel Economy (miles per gallon)

σ = Fraction of Fuel Used to Heat Gas Stream

Equation 33 is plotted in Fig. 29 for typical conditions as specified. Burn times in the region of 2 to 15 minutes are available that limit peak temperature rises to less than 1000°F. Thus, heavier structures are preferred.

Another desirable characteristic of design is that during burn it is desired to have a poor gas-to-structure thermal contact in order to minimize the heat wasted in "soaking" the structure. During the cooling phase, however, the reverse is true. We desire a good gas-to-structure thermal contact in order to "flush" the heat out efficiently.

BUREAU OMPI WIPO INTERNATION

The cooling effects during the non-burn period are calculated next. Things are different than those during burn since the entire energy source is that stored in the structure. The structure will cool from the peak temperature reached at the end of combustion to the gas temperature existing out of the compressor. The cool gas stream out of the compressor will rise initially as the heat is flushed out but then will gradually approach the original value. This situation is shown in Fig. 30. To start, the following definition is made:

$$34. \quad \dot{Q}_a = h \dot{W}_a (T_s - T_o)$$

$\dot{Q}_a$ = Heat Take up of the Air (Btu/minute)

$\dot{W}_a$ = Mass Flow Rate of Air (Pounds/minute)

T_o = Compressor Outlet Temperature (°F)

h = Heat Transfer Coefficient (Btu per Pound - °F)

The heat transfer coefficient is easy to visualize, easy to measure, and fits in well with classical concepts. High values of "h" are desirable since this means a short cooling period which in turn eases the flywheel problem. Many factors that will help determine "h" are under the control of the designer:

1. The mass flow rate through the compressor can be increased or decreased during cooling as dictated by heat removal or overall economics.
2. Amount of turbulent flow as the compressed air passes through the engine during the cooling phase.

BUREAU OMPI WIPO INTERNATIONAL

3. Extent and nature of surface films existing on the engine surfaces in contact with the cooling air.
4. Cooling air residence time.
5. Water injection during cooling can be very cost effective.

The total amount of heat stored in the engine structure is:

35. $\Delta Q_s = 0.15\ W_s\ (T_s - T_o)$

Differentiation of equation 35 gives:

36. $\dot{Q}_s = 0.15\ W_s \frac{dT_s}{dt}$

Equation 36 expresses the rate of heat leaving the structure and must equal the negative of the heat take up by the air from equation 34, thus:

37. $0.15\ W_s \frac{dT_s}{dt} = -h\ \dot{W}_a\ (T_s - T_o)$

Equation 37 integrates to give:

38. $T_s = T_o + [(T_s)_{max} - T_o]\ e^{-t/\tau_c}$

$$\tau_c = 0.15 \frac{W_s}{h\ \dot{W}_a}$$

$(T_s)_{max}$ = End of Burn Structure Temperature

Differentiation of equation 38 gives:

39. $\left(\frac{dT_s}{dt}\right)_c = -\left[\frac{(T_s)_{max} - T_o}{\tau_c}\right] e^{-t/\tau_c}$

$$= \text{Structure Cooling Rate}\left(\frac{°F}{\text{minute}}\right)$$

BUREAU OMPI WIPO INTERNATION

Inspection of equation 34 shows a close similarity to a typical specific heat situation with a changing mass. A good estimation for "h" is therefore around 0.25 Btu per pound - °F, the specific heat of air. This value of h is used in the plot of τ_c shown in Fig. 31. A cooling time constant of less than a minute can be used.

Fig. 32 shows a typical structure temperature history with the burn time as a variable. Fig. 33 shows the same data except with structural thermal mass as the parameter. The heat-up portions are "worst case" and pessimistic for several reasons: The structure temperature will actually tend to level off as the hot gas temperature is approached; a perfect and instantaneous heat transfer path from the gas to the structure was assumed. If, for example, an extremely high air-to-fuel ratio were used, then a low gas temperature would result in much less heat transfer to the structure.

It is an important aspect of this invention to achieve a high gas temperature for efficiency purposes while exposing the structure to a much lower average temperature. It is useful to summarize the key relationships that will allow gas and structure temperatures to be calculated and plotted:

$$26. \quad (\Delta T)_g = 86{,}700 \frac{\mathcal{I}}{\epsilon}$$

$$27 \left(\frac{dT}{dt}\right)_s = 14{,}251 \frac{S\ (1 - \delta)}{E\ W_s}$$

$$28. \quad T_s = T_o + \left[(T_s)_{max} - T_o\right] e^{-t/\tau_c}$$

$$\tau_c = 0.15 \frac{W_s}{h\ \dot{W}_a}$$

$$P = 127{,}600 \frac{S}{E} \left(\frac{\text{Btu}}{\text{hour}}\right) \times \frac{1 \text{ h.p.}}{2545 \text{ Btu/hour}}$$

$$29. \quad P = 50.1 \frac{S}{E}$$

BUREAU OMPI WIPO INTERNATIONAL

The following conditions are assumed for the present calculation:

γ = 0.9 = Fraction of Fuel Energy to Gas Stream

S = 60 m.p.h. = Vehicle Speed

E = 30 m.p.g. = Fuel Economy

$(T_s)_{max}$ = 1200°F = Maximum Structure Temperature

T_o = 600°F = Compressor Outlet Temperature

$\dot{W}_a$ = 30 pounds/minute = cooling air rate

h = 0.25 Btu/pound -°F = Coefficient

ϵ = 35 = Air to Fuel Mass Ratio

P = 100.3 horsepower

This case was calculated and is plotted in Fig. 34. Two points should be noted: The gas burn temperature is approximately 2200°F for Carnot efficiency purposes yet the average structure temperature is held to about 1000°F. The second point is that this "beat Carnot" feature requires only about 0.4 minute (24 second) cool periods every 5 minutes.

Some fraction of the input energy will end up as rotational kinetic energy in the rotating member. This aspect is discussed below with respect to Figs. 60-62.

Horsepower levels have not been stated explicitly but are implied in the various equations. Thus,

BUREAU OMPI WIPO INTERNATIONAL

$$P = 127{,}600 \frac{S}{E}\left(\frac{Btu}{hour}\right) \times \frac{1\ h.p.}{2545\ Btu/hour}$$

$$29.\quad P = 50.1 \frac{S}{E}$$

P = Engine Horsepower

S = Vehicle Speed (m.p.h.)

E = Fuel Efficiency (m.p.g.)

SUMMARY OF DERIVED PERFORMANCE RELATIONSHIPS

It is desirable to have, in one spot, all the relationships useful in estimating performance together with definitions and units

$$(\Delta T)_g = 86{,}700 \frac{\sigma}{\epsilon}$$

$$\left(\frac{dT}{dt}\right)_{sb} = 14{,}251 \frac{S\ (1-\sigma)}{E\ W_s}$$

$$= 284.5 \frac{P\ (1-\sigma)}{W_s}$$

$$T_{sc} = T_o + \left[(T_s)_{max} - T_o\right] e^{-t/\tau_c}$$

$$\tau_c = 0.15 \frac{W_s}{h\ \dot{W}_a}$$

$(\Delta T)_g$ = Gas Temperature Rise During Burn (°F)

σ = Fraction of Total Heat into Gas Stream

ϵ = Air to Fuel Mass Ratio

$\left(\frac{dT}{dt}\right)_{sb}$ = Structure Temperature Rise, Burn (°F/min)

S = Vehicle Speed (m.p.h.)

E = Fuel Economy (m.p.g.)

W_s = Structure Thermal Mass (Pounds)

P = Engine Power (Horsepower)

T_{sc} = Structure Temperature During Cooling (°F)

T_o = Compressor Exit Temperature (°F)

$(T_s)_{max}$ = Maximum Structure Temperature (°F)

τ_c = Structure Cooling Time Constant (Minutes)

h = Heat Transfer Coefficient (Btu/lb.-°F)

BUREAU OMPI WIPO INTERNATIONAL

The preferred configuration can best be stated by the typical example as plotted in Fig. 33:

Vehicle Speed = S = 60 m.p.h.

Gasoline Usage = E = 30 m.p.g.

Fraction of Energy of Combustion Used to Heat Working Gas = δ = 0.9

Maximum Structure Temperature = $(T_s)_{max}$ = 1200°F

Compressor Exit Temperature = T_o = 600°F

$\dot{W}_a$ = Cooling Air Rate = 30 Pounds/Minute

Heat Transfer Coefficient Describing Effectiveness of Cooling Air in Flushing the Engine = h = 0.25 Btu/Pound -°F

Air to Fuel Mass Ratio = ϵ = 35

Engine Horsepower = P = 100.3 h.p.

Burn Period = 5 minutes

Cooling Period = 0.4 Minutes

Thermal Mass of Engine = 30 Pounds

The available Carnot efficiency for this example is:

T_{max} = 2225°F = 2685 °R (From Fig. 33)

T_{ex} = 100°F = 560°R (Hot Day)

n_{max} = 79%

Other examples can be calculated as required from the given equations.

BUREA OMPI WIPO INTERNATIC

The following aspects of this invention should be noted:

1. The critical materials of construction; the combustion chamber, ducts, and turbine wheel must be able to withstand repeated heat/cool cycles. An engine lifetime of 100,000 miles at an average of 25 miles per hour will require a total of 48,000 complete cycles at 5 minutes per cycle, for example. The high temperature of the structure is, in one embodiment, around 1200°F and the low temperature is 600°F. The material will be in contact with gas at 2200°F and the surface therefore must withstand this exposure.
2. The combustion chamber, ducts, and turbine wheel must have as high a thermal mass as possible. That is, it must take a long time to heat up.
3. The gas turbine of this invention allows a maximum amount of heat to enter the gas stream and a minimum amount of heat to enter the engine structure.
4. The engine characteristics during cooling should be the reverse, i.e., heat removal via the cooling air stream should be rapid.
5. The amount of compressed air used during cooling may be more or less than that used during burn. This compressed air is not wasted since it does pick up heat and it does supply energy to the turbine.

BUREAU OMPI WIPO INTERNATIONAL

6. Various techniques can be used to impede heat transfer to the structure during burn and aid heat removal from the structure during cooling. For example, water injection during cooling can be used. The heat of vaporization of water can rapidly cool the engine. The steam so generated can be used to drive the turbine.

7. The cool period should be as short as possible in order to minimize the flywheel problem.

8. Throttle control can be accomplished three ways: (1) vary the time of burn, (2) vary the time between burns, and (3) vary the fuel during burn.

9. The gas pressure required during cooling can be much less than during burn. Thus, a "by pass" can be used during cooling where great amounts of ambient air are purposely routed around the compressor through the hot engine.

10. The intermittent operation concept improves efficiency at all throttle levels high and low. The classical poor efficiency at part load due to low combustion temperature no longer exists.

11. A closed recycle system can be used. during cooling, for example, the steam from a water injection system can be condensed and recycled.

BUREAU OMPI WIPO INTERNATION

12. The high combustion temperatures allow for a greatly reduced air/fuel ratio (Fig.28). This has three major advantages: the volume of the exhaust and intake gas is reduced, high exhaust temperature is less a problem since less total energy is involved, and compressor requirements are reduced.

A simple turbine engine is shown in Fig. 35. The schematic symbols used in Fig.35 are used throughout this specification and will be clearly understood by those skilled in the gas turbine field. Reference numerals 310 and 312 represent the compressor and turbine, respectively, and any well-known types of compressors and turbines can be used. The lines 316, 318, 320, 322, 328, and 330 schematically represent the flow of air and of exhaust gas out of the gas turbine. Fuel line 324 and water injection line 326 are self-explanatory. The control means 315 mixes the fuel and air burn mixture (and any other ingredient thereof), and programs the burn/cool cycle, and also provides general control, as will be well understood by those skilled in this art and therefore no detailed description thereof is necessary or desired. The control means 315 preferably contains a microprocessor.

BUREAU OMPI

SPECIFIC GAS TURBINE CONFIGURATIONS

The following portion of this specification describes several specific structural configurations with reference to Figs. 36-59.

The piston engine is todays prime mover and many configurations are sold to satisfy many requirements. The same situation is true for the gas turbine. Gas turbine engines in large luxury cars differ greatly from those in compacts. The relative importance of economy and pollution are not the same in Japan as in the U.S. Thus, gas turbines for service in the U.S. differ from those in Japan. The gas turbine for truck service, golf cart service, railroad locomotive service, and for general power station service are different.

Quantitative numbers for the configurations below, are found in other portions of this specification. The intermittent operation of the present invention is assumed throughout and in most instances a flywheel of some type is used.

A review of some basics applicable to all configurations will first be set forth. Main functions are: Compress a gas (typically ambient air), expand a compressed/hot gas to do useful work (turbine), heat exchangers, and control. The control, burn box is not actually one physical box. This function includes mixing fuel with compressed air, ignition, programming an appropriate burn/cool period, route and re-route air and gasses, accept throttle input data, etc. A microprocessor is preferred for use in the control means. The normal housekeeping functions of burn control, response to throttle, etc. will be assumed

BUREA OMPI WIPO INTERNATI

and are thus not necessary to be discussed for each configuration. One other basic fact that may be useful to be reviewed is as follows; Fig. 36 represents a simple, single shaft turbine. Ambient air is sucked into the compressor and passed through the combustion chamber. The hot, compressed gas stream passes through the turbine where expansion makes the shaft turn at high speed. The turning shaft is made to perform two functions: turn the compressor and thus supply the necessary compressed air and turn the output shaft which may drive a gear box, differential, etc. Turbine exhaust energy mostly in the form of heat may or may not be recovered. Fig. 49 shows a typical two shaft or twin turbine engine. The first turbine 312 drives the compressor 310 only. The exhaust gas from the first turbine 312 contains residual pressure and is used to drive the second, or power turbine 376. The advantage of a two shaft device is that the two turbines 312 and 376 can run at different r.p.m.s. In general, the compressor 310 requires a high r.p.m. for good efficiency but the load requires a much lower r.p.m. The two-shaft unit thus can eliminate the need for gear reduction boxes.

Referring now to Figs. 36-59, Fig. 315 shows a simple, single shaft gas turbine including a compressor 310, turbine 312, shaft 314, flywheel 332, and control means 334 (the box 334 also schematically representing the burn chamber). The simple mechancial flywheel 332 is provided to supply a smooth continuous output motion from the intermittent burn histroy.

Fig. 36 also shows a surge tank 331 connected between the compressor 310 and turbine 312. The tank 331 is kept full of compressed air from the compressor 310. The output of the tank 331 is controlled by a valve 333, the state of which is controlled by the control means 334.

BUREAU OMPI WIPO INTERNATIONAL

The valve 333 can be opened and the compressed air in the tank 331allowed to drive the turbine 312 for a short period of time to provide a sudden burst of power when needed, for example.

Fig. 37 also shows a single shaft engine but includes an electrical flywheel instead of a mechanical one. An electric generator 336 is mechanically coupled to the turbine shaft 314. The generator output, after any appropriate processing such as rectification to D.C., is used to charge an electrical energy storing means such as a capacitor or battery 338 during the "burn" period. During the "cool" period, the charged battery 338 drives the generator 336 as a motor. This action is automatic and is similar to the dynamic braking used in railroad service.

Fig.38 shows an example of a chemical flywheel. A cryogenic "fuel" such as dry ice, liquid air, or liquid oxygen is vaporized and passed through the system during the cooling period. The cryogenic fuel serves two purposes: it expands and drives the turbine 312 and it cools all the exposed parts that were heated during the burn cycle.

Fig.39 shows a simple, single shaft engine where more (or less) air is passed through the system during cooling than during burn. The quantitative numbers for this embodiment are given in another section of this specification.

BUREAU OMPI WIPO INTERNATIONAL

Fig. 40 shows a single shaft engine employing a mechanical flywheel 332 and water injection. The main purpose of the water injection is to expedite heat transfer out of the engine during the cooling phase.

Fig. 41 shows a single shaft engine and the use of more than one fuel. One fuel, such as oil or gasoline, is used during the burn phase and a second fuel is used during the cool phase. The configuration of Fig. 38 is a special case of this embodiment of Fig. 41. The flywheel 332 can be dispensed with in this configuration, if desired.

Fig. 42 shows a modification of Fig. 40 wherein the injected water is recovered in a condenser 352 and recycled.

Fig. 43 shows a single shaft engine that includes an electric generator 336 mechanically coupled to the shaft 314. The generator output is used to drive an electric traction motor 354. A mechanical flywheel 332 is also included.

Fig. 44 shows a modification of Fig. 44 with the addition of a gear box 358 to mechanically couple the high speed turbine 312 and the lower speed electrical generator 336. A mechanical flywheel 332 is included. Belts can be used in place or in addition to the gear box, as desired.

Fig. 45 shows an engine including an electric generator 336 plus an electric traction motor 354. In addition, the generator simultaneously charges a capacitor or battery 38 to provide a flywheel action. Note also that dynamic braking, similar to standard railroad practice, is also supplied. As the vehicle slows down, the motor 354 becomes a generator and the generator becomes a motor (automatically). The net result is a braking action plus the vehicle kinetic energy ends up as stored charge on the capacitor. This results in high fuel savings in stop and go traffic. The burn, control box 362 assures that operation at all times keeps the capacitor 338 (or battery) charged.

BUREAU OMPI WIPO INTERNATIONAL

Fig. 46 is a generic arrangement that allows the control means 364, the motor 354, generator 336, and battery 338 to communicate with each other, for the greatest flexibility of operation.

Fig. 47 shows a single shaft turbine with electric drive and electric flywheel action wherein waste heat in the exhaust gas is recovered in heat exchanger 368. The hot exhaust gas is used to pre-heat the compressed air thus saving fuel.

Fig. 48 shows a preferred single shaft, electric drive system for large car or truck service. Water injection is shown and the water is recovered and recycled.

The above-described embodiments are single shaft arrangements. Many additional combinations can be used, such as a mechanical flywheel with water injection, water recovery, and exhaust heat utilization.

The following embodiments are multiple shaft concepts.

Fig. 49 shows the simplest of the twin shaft approaches. A mechanical flywheel 332 on the compressor/turbine shaft 314 is provided. The output or load shaft 378 can drive a gear box, traction wheels, compressor. etc.

Fig. 50 shows a modification of Fig. 49 wherein a mechanical flywheel 332 is attached to both turbine shafts 314 and 378. The flywheels 332 can differ from each other and can be considered as devices to provide a desired moment of inertia for the two shafts.

Fig. 51 shows a two shaft embodiment where the compressor shaft 314 has an electric flywheel (electric generator 336 and battery 338) and the power shaft 378 has a mechanical flywheel 332.

Fig. 52 shows a combined cycle. The hot exhaust gas from the power turbine 376 is used to vaporize a second working fluid in a closed turbine system. The vapor drives the compressor turbine 312 only. The fluid is condensed at 372 and recycled in a closed system. The working fluid can be water, ammonia, freon, Learium, etc.

BUREAU OMPI WIPO INTERNATIONAL

Fig. 53 is similar to Fig. 52 except for a mechanical flywheel 332 in place of an electrical flywheel.

Fig. 54 shows a similar combined cycle except that the compressor turbine 312 is gas driven and the output turbine 376 is in the closed system.

The embodiments of Figs. 52-54 can be open loop on the secondary loop if plenty of water is available on board the vehicle (for example, a locomotive).

Fig. 54 shows a twin shaft engine with an electrical generator 336 on each shaft 314 and 378. The generators can communicate with each other and can charge or draw energy from each others' battery 338. As in all electric schemes described above, the electric power generated can be used for auxiliary purposes in addition to traction, such as lights, heaters, power steering pumps, etc.

Fig. 56 shows a combined cycle on a single shaft with an electric flywheel and electric drive. Both mechanical work (output shaft) and electrical work via the traction motor are available.

The various components of each of the above systems are well-known to those skilled in the art and any appropriate known type of available component can be used in this invention. For example, one standard technique used today that can be useful in this invention is the compact, concentric design of twin shaft units where one shaft is hollow and the other shaft lies inside of the hollow shaft.

The fuel management and throttle aspects are discussed in more detail elsewhere. For convenience, in the drawings, the throttle input is shown at the burn, control box. Actually, a signal will appear there and at the power turbine simultaneously. A detailed discussion is given elsewhere in this specification.

BUREAU OMPI WIPO INTERNATIONAL

The embodiments shown in Figs. 57-59 operate to lengthen the cooling period while at the same time reducing the flywheel requirements.

Fig. 57 shows a four turbine configuration including turbines 312, 312a, 312b, and 312c. The corresponding burn cycle chart is shown in Fig. 58. Clearly, the flywheel requirements become very minimal and must only supply continuity during the short burn change-over from one turbine to the next. The cooling periods are much greater than the burn periods and very hot, and short, burn periods can be used. The net result is a relatively small engine due to the low mass flow required (see the earlier discussion). These advantages are at the expense of added complexity. Any number of time-shared turbines can be used.

It is to be understood that the arrangement shown in Fig. 57 can use any and all of the features described above with respect to Figs. 36-56, for example, such as the electric flywheel, electric drive traction motor, water injection, heat exchange between exhaust and inlet air, etc.

The multi-turbine, time-sharing feature of this invention can also be applied to a free turbine. Fig. 59 shows such an embodiment for a 3-compressor turbine/3 power turbine arrangement. This free turbine feature can also utilize any and all of the features described above, for example, in Figs. 36-56, electric drive, electric flywheel, water injection, etc.

For a sudden burst of speed, the multi-turbine arrangement has the advantage that you can "fire-up" all of them (or several) at the same time. For non-plural turbines, gas can be compressed and stored and fed to the turbine (or to the free turbine, if used) to provide a sudden burst of power. It is not essential that such gas also be burned. At such time, if a mechanical flywheel is used it can be decoupled from the shaft, as by a fast acting clutch.

BUREA OMPI WIPO INTERNATIC

Further, since this embodiment provides such a long cool period (compared to what is needed), as shown in Fig. 58, another cycle that can be used is that in which the fuel burn phase is represented by the open areas in Fig. 58 and the cool phase is shown by the closed areas. That is, instead of burning for 1/4 of the cycle, the burn would be for 3/4 and the cool for 1/4 of the cycle, for each turbine and burn chamber.

It is noted that Fig. 57 provides a "parallel" (as contrasted to prior art "series") configuration using a plurality of turbines and burn chambers.

THE FLYWHEEL

The gas turbine of the present invention is based upon intermittent operation. Fuel burn periods of 5 minutes duration are typical. Cooling periods of about 30 seconds duration are interspaced between burns. Some type of flywheel action is desired in order to assure uniform and continuous output power.

The flywheel can be mechanical, electrical, or chemical in nature. Mechanical flywheels are familiar and well-known and can be used. However, modern high strength-to-weight plastic materials can also be used. An electric flywheel can also be used. Consider an electric generator that is mechanically coupled to the turbine shaft. The generator output is used to charge an electrical capacitor or battery during the power or burn phase. During the cool phase the electric generator will automatically become a motor and the capacitor or battery will drive it accordingly. This is the same "dynamic braking" action that is common railroad practice. One method of "chemical flywheel" practice is to inject a cryogenic material during the cooling period. The vaporization and expansion of this material will not only cool the engine interior but will also drive the turbine.

BUREAU OMPI WIPO INTERNATIONAL

The mechanical and electrical flywheels are discussed below, and quantitative data is derived and presented. All cryogenic aspects of the turbine of this invention are discussed in a separate section below.

Regarding now the mechanical flywheel, typical automotive cruise power levels are of the order of 50 horsepower. Burn periods of 5 minutes and cooling periods of 0.5 minutes dictate that the flywheel must be able to store about 0.5 x 50 = 25 horsepower minutes worth of energy.

Consider the simple flywheel shown in Fig. 60. We can write:

1. $E = \frac{W}{32.2} R^2 \omega^2$

E = Energy of Rotation (Foot Pounds)

W = Total Wheel Weight (Pounds)

R = Wheel Radius (Feet)

ω = Rotational Speed (Radions/Second)

Also, we can write:

2. $V_t = R\omega$

V_t = Tip Speed (Feet/Second)

A limit of unity mach number exists due to practical limitations. It is convenient to write:

3. $V_t = 1000\alpha \qquad 0 \leq \alpha \leq 1.0$

Since for $\alpha = 1$, V_t = Mach 1,

It follows that:

4. $R\omega = 1000\alpha \qquad 0 \leq \alpha \leq 1$

Insertion of equation 4 into equation 1 gives:

$E = 3.11 \times 10^4 \alpha^2 W$

1 h.p.-sec. = 550 Foot-Pounds, thus

E (h.p.-min.) = $0.94\ W\alpha^2$

W = Wheel Weight in pounds.

BUREAU OMPI WIPO INTERNATIONAL

The December 1973 issue of "Scientific American" contains an article on the design of flywheels using modern materials. The article was on the use of flywheels to supply primary power to an automobile. Thus, such a flywheel must store, for example, 100 h.p.-hours as opposed to 100 h.p.-minutes in the present invention. Thus mechanical flywheels that can be used in the present invention are within current state-of-the-art, and therefor no detailed discussion thereof here is necessary or desired. Fig. 61 charts the flywheel stored energy versus its weight, at several values of tip speed.

Regarding the electrical flywheel, the equipment required here is not much more complicated than a D.C. generator plus a capacitor or battery. An A.C. generator (alternator) plus silicon rectifiers will also satisfy the requirements. The energy storage can be either a battery or a capacitor. It is very important to note that the requirements for such an electrical flywheel for this invention are much less stringent than for an electric vehicle. The requirements in the present invention dictate that the battery or capacitor must drive the vehicle for seconds and not hours.

Typical lead-acid batteries will store about 4.0 h.p.-minutes per pound of battery. Thus:

7. $E_b = 4.0\ W_b$

E_b = Battery Stored Energy (h.p.-min.)

W_b = Battery Weight (pounds)

Equation 7 is plotted in Fig. 62 and very clearly a 50 pound battery can satisfy the requirements.

An electrical capacitor can also be used. The energy stored in an electrical capacitor is:

BUREAU OMPI WIPO INTERNATIONAL

8. $E_c = \frac{1}{2} CE^2$

E = Capacitor Energy (watt-sec.)

C = Capacitance (Farads)

E = Applied Potential (volts)

A change of units gives:

9. E (h.p.-min.) $= 1.11 \times 10^{-5} CE^2$

Electrolytic capacitors give the best storage density today. The most optimum conditions find:

10. C = 2000 microfarads

E = 450 volts

W_c = 1 Pound

Insertion of equation 10 into equation 9 gives:

11. E_c (h.p.-min.) $= 4.5 \times 10^{-3} W_c$

W_c = Weight of capacitor (Pounds)

Comparison of equation 11 to equation 7 shows the battery to be superior to the capacitor by 1000:1 on a per weight basis. For this reason, the capacitor data will not be plotted. It should be noted, however, that the capacitor energy storage density varies with the square of the dielectric strength. Thus, a 30:1 improvement in dielectric strength would allow the capacitor to compete with the battery on a weight basis.

Summarizing, a mechanical flywheel and a battery driven electric flywheel can both satisfy present requirements. The mechanical flywheel is simple and maintainence free. It does, however, present a minor safety problem in case of an accident.

A battery powered electric flywheel is more complicated than its mechanical counterpart. However, the electric version adds almost no cost if an electric traction option is adopted. Battery lifetime is of some concern depending upon how deep the discharge is each cycle. A capacitor approach must await a 30:1 improvement in dielectric strength. The capacitor enjoys simplicity and long life.

BUREAU OMPI WIPO INTERNATIONAL

EXHAUST ENERGY LOSS REDUCTION

The exhaust stream power loss, for a fixed temperature, is directly proportional to the mass flow rate through the turbine engine. In addition, the exhaust management and control problems tend to increase in severity at high exhaust rates and temperatures.

The orthodox attack taken on this problem by the auto industry is two-fold:

1. Develop materials that will tolerate a higher combustion temperature. This process will automatically yield lower exhaust gas volumes via reduced air-fuel ratios.
2. Develop better heat exchangers that couple the hot exhaust gas stream to the cooler inlet gas stream. This action will both reduce the exhaust temperature and reduce the fuel requirements.

The intermittent operation of the present invention is another approach to the exhaust problem. This invention allows for significant increases in combustion temperatures which greatly reduces the air-to-fuel ratio. The net result is a marked reduction in exhaust gas volume and power loss. This air/fuel ratio reduction can only be carried so far before emissions rich in undesirable nitrogen oxides appear. The air-to-fuel ratio in the prior art turbine is around 50:1. The natural burn ratio for gasoline is about 15:1 and emission problems disappear above 17:1. Essentially, the higher ratios supply more than adequate oxygen for combustion which tends to minimize the oxidation of nitrogen. The intermittent operation of the present invention calls for air to fuel ratios of from about 14.7:1 to 35:1.

BUREAU OMPI WIPO INTERNATIONAL

The combined cycle discussed above can reduce exhaust losses further still. The hot exhaust gas stream is used to vaporize an auxiliary working fluid which in turn drives another (or the same) turbine.

The following analysis provides quantitative relations in regard to exhaust energy management.

It is useful to extract certain results from the discussion above regarding Figs. 22-35. The fuel is assumed to be gasoline throughout the following analysis. From the above-mentioned earlier discission, we can write:

1. $Q_g = 127,600 \frac{Btu}{gal.} = 20,750 \frac{Btu}{pound}$

2. $(\Delta Ta)_b = 86,700 \frac{\alpha}{\epsilon}$

3. $P = 50.1 \frac{S}{E}$

4. $\dot{Q}a = 0.24 \dot{W}a \Delta Ta$

5. $P = 489 \dot{W}_f = 489 \frac{\dot{W}a}{\epsilon}$

Where:

Q_g = Heat content of gasoline

$(\Delta Ta)_b$ = Temperature rise of air stream (°F)

α = Fraction of heat used to heat gas

ϵ = Air to fuel mass ratio

P = Vehicle Power (horsepower)

S = Vehicle Speed (miles/hour)

E = Fuel Economy (miles/gallon)

$\dot{Q}a$ = Heat rate in air stream (Btu/minute)

$\dot{W}a$ = Mass flow rate of air (pound/minute)

ΔTa = Temperature rise of air (°F)

$\dot{W}_f$ = Fuel mass flow rate (pound/minute)

Equation 4 can be written:

6. $Pa = 5.66 \times 10^{-3} \dot{W}a \Delta Ta$

Pa = Power in air stream (horsepower)

Insertion of equation 5 into equation 6 yields:

7. $\left(\frac{Pa}{P}\right)_{\%} = 1.16 \times 10^{-3} \epsilon \Delta Ta$

Equation 7 expresses the percentage loss in the exhaust stream as a function of air-to-fuel ratio and gas stream temperature. The results are plotted in Fig. 63.

BUREAU OMPI WIPO INTERNATIONAL

Assuming the use of a fuel-to-air ratio in the present invention of 25:1, Fig. 63 shows the exhaust loss to only be 15% of the total power for an exhaust temperature of 550° F. This compares to a 46% loss at a fuel to air ratio of 75 (typical of the prior art engine).

In short, the intermittent operation of the present invention goes a long way in solving the exhaust temperature loss problem. This result is an unexpected bonus of the high gas temperature made possible by the intermittent operation.

THERMODYNAMIC CYCLE SELECTION

The gas turbine engine of the present invention operates in a continuous but interrupted mode; i.e. an intermittent burn cycle comprising a burn phase followed by a cool phase. The principle of operation can best be visualized by considering a fixed throttle and constant load case. Ambient air is compressed, mixed with a vaporized fuel, burned, and the hot, compressed gas stream is allowed to continuously expand and rotate a work turbine. This action should be smooth and continuous and the work turbine can be made to drive an electrical generator or to move a mechanical transmission device. During this period of fuel burn (the burn phase), the intermittent operation of the present invention draws heavily upon orthodox gas turbine technology. However, during the "cool" phase, the fuel flow is interrupted and little or no fuel is fed to the burn chamber for brief periods of time in order to limit the peak and average engine structure temperatures to acceptable values. The period of interruption must be less than the thermal time constant of the engine structure. The duration of fuel cut-off need only be seconds rather than minutes due to the excellent heat exchange properties of the high volume of cool, compressed ambient air passing through the engine. Exact cycle parameters are variable but 5 minute burn periods and 30 second cooling

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periods are typical. In multi-turbine configurations, a 1 minute burn and 6 minute cooling period may be used.

The engine is structured such that no more than 10% of the heat from the burning fuel is "soaked" into the structure. It follows that total thermal efficiency is determined primarily by conditions existing during the burn period. Nevertheless, the heat flushed out of the engine structure during cooling does provide useful work against the turbine. In any case, it is important to select the proper thermodynamic conditions during the burn phase. Many thermodynamic cycles are possible: Carnot, Diesel, Otto, Stirling, Erricsson, Brayton (Joule), etc. However, the Brayton can best satisfy the continuous and high speed requirements demanded by the turbine. However, if a burn-to-cool duty cycle much different than the presently preferred 90%/10% is used then a cycle other than Brayton's can be used.

THE BRAYTON CYCLE ADAPTED TO THE INTERMITTENT OPERATION OF THE PRESENT INVENTION

The ideal (or reversible) Brayton cycle assumes that the compression and expansion processes will waste no energy. The practical or irreversible Brayton cycle recognizes that losses do occur in compression and expansion processes. The ideal cycle is easy to analyze and the practical cycle is difficult.

An understanding of the ideal Brayton cycle serves no end use in itself but it does expedite an understanding of the practical cycle. The ideal cycle is described by the pressure, volume (p,v) and the temperature, entropy (T,S) curves in Fig. 64. The principle of operation can best be related to the four circumscribing paths 1-2, 2-3, 3-4, and 4-1.

1-2 A unit mass of air is adiabatically compressed from ambient pressure P, to a higher pressure P_2. The process is reversible and the entropy is constant since no heat is lost.

2-3 Heat is added (fuel burn) and the hot gas stream is allowed to expand at constant

BUREAU OMPI WIPO INTERNATIONAL

pressure. The entropy and temperature are both increased.

3-4 The hot, compressed gas stream is allowed to reversibly and adiabatically expand against the work turbine. There is no entropy change but temperature decreases, pressure decreases, and volume increases.

4-1 A volume of hot gas is exhausted to atmosphere at fixed pressure (atmospheric). The temperature, entropy, and volume of the working fluid decrease while the pressure remains constant.

If no attempt is made to recover the waste heat, then the thermal efficiency can be written: (see p. 44 of "Small Gas Turbines")

1. $(\eta_b)_I = 1 - \frac{P_i}{P_2}^{0.285}$

$(\eta_b)_I$ = Efficiency of Ideal Brayton

Subscripts refer to Fig. 64, thus P_2 is pressure at point-2. An important fact to note is that the ideal Brayton cycle efficiency depends only upon the pressure ratio if no attempt is made to recover the waste heat. A similar result follows (see p. 477 of "Handbook of Tables for Applied Engineering Science") for the gasoline piston engine (Otto) and the two results are plotted in Fig. 65. The gasoline engine operates at about a 50:1 pressure ratio and the Brayton at perhaps 5:1. The gasoline (ideal) is seen to be vastly more efficient. However, in the practical case, the Brayton will win out since it is more amenable to exhaust heat recovery.

An expression for the thermal efficiency of a practical Brayton cycle (see p. 102 of "Theory and Design of Steam and Gas Turbines").

2.

$$\eta = \frac{(P^{0.3} - 1)\left(0.85\, P^{-0.3} - 1.176\, \frac{T_1}{T_3}\right)}{1 - \eta_R\left[1 - 0.87\,(1 - P^{-0.3})\right] + \frac{T_1}{T_3}(1 - \eta_R)\left(\frac{P^{0.3} - 1}{0.87} + 1\right)}$$

BUREAU OMPI WIPO INTERNATIONAL

P = Pressure Ratio

T_i/T_3 = Absolute temperature ratio of ambient to combustion

η_R = Efficiency of Regenerator

Reasonable values of compressor and turbine mechanical and work efficiencies are assumed. Equation 2 is plotted in Fig. 66 and the advantage of high combustion temperatures is clear. A combustion temperature of 2500° F and an 90% regenerator yields a net thermal efficiency of about 43% and represents a significant improvement over the gasoline/piston engine.

Judge (see p. 90, Fig. 45 of "Small Gas Turbines") presents similar data that was taken from a real engine and is considerably more optimistic. One curve is included in Fig. 66. Judge (p. 58, Fig. 25 of "Small Gas Turbines") gives another curve that relates total engine thermal efficiency vs. combustion (turbine inlet) temperature for a real engine. This curve is reproduced here as Fig. 67. A special high temperature alloy utilizing hollow, air-cooled turbine blades were required in order to obtain the data. Typical diesel efficiencies are given. It is clear that the 2500° F (approx) gas temperature in the present invention surpasses the diesel in efficiency.

FUEL INJECTION, MANAGEMENT AND CONTROL

Two long standing efficiency problems cause the prior art gas turbine engine to fall short of its ultimate potential:

1. It is well established that the prior art engine suffers severly in performance at

BUREAU OMPI WIPO INTERNATIONAL

reduced load. The reason is clear. A high thermal efficiency requires a high combustion temperature by the Carnot principle. Reduced load throttling in the prior art gas turbine is achieved by reducing the fuel burn rate (and also the air-to-fuel ratio). The net result is a low combustion temperature since the air mass flow rate must remain essentially unchanged during the throttling process.

2. The maximum achievable average combustion temperature is severly limited by the availability of temperature resistant materials.

The intermittent cycle of the present invention solves both of these problems completely. A high combustion temperature (for efficiency) is used simultaneously with a lower engine average temperature as a result of the intermittent burn-cool-burn-cool cycle. The reduced load problem ceases to exist since throttling is accomplished by changing the length of the burn and cool periods (rather than by reducing the amount of fuel fed into the combustion chamber). The fuel burn rate in the present invention, and hence the combustion temperature, is always held at the same fixed maximum value, during maximum conditions.

The prior art gas turbine engine problems can be briefly stated with reference to Fig. 68. The horsepower setting, the combustion temperature, and the fuel rate all vary with the throttle setting. A simple graph of the throttle situation of the present invention is shown in Fig. 69.

From Fig. 69 we can write:

BUREAU OMPI WIPO INTERNATIONAL

$$P_{aug} = P_p \left[\frac{t_B}{t_B + t_c} \right] \quad \text{or,}$$

$$1. \left(\frac{P_{avg}}{P_p} \right) = \left[\frac{1}{1 + t_c/t_B} \right]$$

P_{avg} = Average Engine Horsepower

P_p = Peak Horsepower During Burn

t_B = Burning Time Interval

t_c = Cooling Time Interval

$\left(\frac{P_{avg}}{P_p} \right)$ = Throttle Setting

The peak power is held constant at all throttle settings and consistent with efficiency requirements. The throttle setting is modulated by adjusting the ratio of the cool-to-burn periods and results in a good flexibility of design. For example, 1 minute of burn followed by 10 seconds of cooling will yield the same percent throttle as 2 minutes of burn followed by 20 seconds of cooling. The absolute values of the burn and cool periods can be determined for each particular application by:

1. The flywheel energy storage capacity.
2. The thermal capacity of the engine.
3. The allowable temperature excursions of the engine.
4. Engine/air heat transfer efficiency during cool phase.

Equation 1 (above) is plotted in Fig. 70 which shows that preferably both the burn and cooling periods are modulated in order to provide desired dynamic range. Consider, for example, a fixed 30 second cool period. A 90% throttle setting demands a reasonable 4 minute burn period, but a 30% throttle demands an undesirably short burn period of 0.2 minutes. Similarly, a 2 minute cool period is adequate at some throttle settings and inadequate at others. In any case, an acceptable combination of burn and cool periods does exist for all throttle

BUREAU OMPI WIPO INTERNATIONAL

settings. A typical accelerating throttle history is shown in Fig. 71.

Different configurations of control apparatus will be used for different applications. A very simple voltage regulation system will be adequate for a gas turbine/electric drive golf cart equipped with a storage battery. However, a more elaborate system would be required for a truck or railroad locomotive.

Fig. 72 shows a typical single shaft turbine utilizing an electric drive. A standard electric generator 336 is coupled to the turbine shaft 314 either by gears, a pulley and belt, or directly. The generator output charges an electric battery 338. If the generator is an a.c. alternator then rectifiers are provided. The battery 338 is connected to a traction motor 354 by a control box 410. Akkutechnik Ladegerate G_{mbH} of West Germany makes a family of such controls specifically for electric drive vehicles. In the November 1977 (page 30) issue of "Electric Vehicle News", their Model "241" is described and it can be used as control box 410, as can any other known and appropriate means, as will be well understood by those skilled in this art. A sensor 412 is used to detect the state of battery charge. Stoneleigh Electronics of Romford, Essex, England advertises an appropriate level sensor in the same November 1977 issue of "Electric Vehicle News", also on page 30. Several actions occur as the battery drops below a present level:

1. Fuel is metered to the combustion chamber 100 via a standard solenoid actuated valve 416.
2. A starting ignition spark is supplied by ignition control means 422.
3. A relay 414 disconnects the battery 338 from driving the generator 336 as a motor. The generator 336 now commences to charge

BUREAU OMPI WIPO INTERNATIONAL

the battery 338.

4. If the engine is started from a cold start, the battery 338 must initially turn the shaft 314 via the generator 336 connected as a motor. This action is initiated by a standard motion sensor 426 or a manual starting switch connected to the motor start relay 424.

An output temperature sensor/switch 428 arrangement is supplied to provide automatic cut-off is case of excessive temperatures. The fuel pump 420 and ignition system can be standard home oil heating components. The entire compressor 310, turbine 312, and combustion system 400 can be supplied by the commercially available units from Solar Aircraft of San Diego. These Units of 75 h.p. (Mercury) and 500 h.p. (Jupiter) are described in "Small Gas Turbines" by Judge. Regarding the various components of the system shown in Fig. 72, these are well-known to those skilled in the art and therefore do not need to be described in detail here. For example, the generator 336 can be such as are made by GM, EMD, CAT, Allis, Blackstone, Cummings, etc., from 2 to 2500 KW in one unit. Further, entire families of DC motors and generators are available such as from GE. The level sense 412 can be units such as made by WAMS, Inc. and such as are used to protect electric forklifts. The battery can be such as are made by Gould for lift trucks. The compressor 310 as well as other components of the gas turbine can be such as are made by Norwald-Turbo, Inc., subsidiary of Union Corporation. Regarding the control, such units are well-known as described in the January 1978 issue of Control Engineering starting at page 62; units are available commercially which provide, for example, the 45 different functions or features set forth at page 63 thereof. Regarding the motor 54, either an A.C. or D.C. motor can be used, all as is

BUREAU OMPI WIPO INTERNATIONAL

well-known in the art, see for example, the January 1978 issue of Control Engineering starting at page 58.

In fact, a very simple implementation of the intermittent cycle of the present invention can use such equipment plus a flywheel 332 and a manual valve 434 as shown in Fig. 73. Fig. 73 shows a fuel tank 432, a valve 434 (which can be manually operated), a gas turbine 430 (such as that shown at page 303 of Judge, "Small Gas Turbines"), a flywheel 332 and a temperature gauge 435. The manual valve 434 can be opened and closed by hand consistent with the demands of the load and the temperature of the structure as shown by the gauge 435 (a temperature of 1100° F should not be exceeded), and the teachings set forth above.

The above description has been with respect to a "steady state" condition of operation. However, to provide instant response to throttle, the system will include an "override", whereby the intermittent cycle of this invention is "switched out" and a sudden change or burst (acceleration or deceleration) of power can be achieved by using the known continuous burn operation.

The following typical numbers may be useful in appreciating the vast amounts of energy used today and how advantageous the increased efficiency of the gas turbine of the present invention is: (1) the automobile uses at maximum 1,250,000 BTU/hr or 10 gal./hour, and at idle 125,000 BTU/hr; (2) a truck uses at maximum 5,000,000 BTU/hr or 50 gal/hour; (3) the airplane and locomotive use about 50,000,000 BTU/hr or 500 gal/hr, and (4) central station power uses 10^{10} BTU/hr or 100,000 gal/hr.

BUREAU OMPI WIPO INTERNATIONAL

The invention has been described in detail with particular reference to the preferred embodiments thereof, but it will be understood that variations and modifications can be effected within the spirit and scope of the invention as described hereinabove and as defined in the appended claims.

Another aspect of the present invention is the throttle control invention described below, which is also described in U.S. application Serial No. 890,465, filed March 27, 1978 entitled "Gas Turbine System" by the same inventor as the present application. While this description is primarily with reference to a Brayton cycle gas turbine, all embodiments thereof are also applicable and can be used in combination with all aspects of the present invention of a positive displacement Brayton cycle engine, and also can be used in further combination with all combinations of the positive displacement Brayton cycle engine with all aspects of the "Gas Turbine System" of application Serial No. 879,969 described above.

SUMMARY OF THE THROTTLE CONTROL ASPECT

A gas turbine method and apparatus using either a continuous burn operation or an intermittent burn operation cycle and including: means for controlling the air-to-fuel ratio in the primary combustion zone, means for maintaining the air-to-fuel ratio at a substantially constant value even as the fuel rate varies, means for maintaining the burn temperature in the primary combustion zone substantially constant even as the fuel rate varies, and means for varying the quantity of the compressed air fed to the secondary zone of the combustion chamber for cooling the turbine, independently of the control of the fuel rate and the primary air fed to the primary combustion zone. The combustion process, during either steady throttle or varying throttle demand, can be varied

BUREAU OMPI WIPO INTERNATION

by varying any one or more of the following parameters: the fuel rate, the length of the burn phase, the length of the cool phase, and the quantity of cooling air fed to the secondary zone to limit the turbine temperature.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

In operation, the embodiments of the present invention can be operated in either the normal, prior art continuous burn mode or in the intermittent burn operation cycle of the Lowther Gas Turbine as described in applicant's copending patent application Serial No. 879,969, filed February 21, 1978, and entitled "Gas Turbine System". Said entire copending patent application Serial No. 879,969 filed February 21, 1978 and Serial No. 890,465, filed March 27, 1978 by applicant are hereby incorporated by reference in this application.

To summarize the disclosure in applicant's said copending patent application Serial No. 879,969, the intermittent burn operation cycle comprises a burn phase (such as of 5 minutes) followed by a cool phase (such as of 30 seconds). During the burn phase fuel is continuously burned in a conventional manner except at a much higher and more efficient temperature. During the cool phase, the fuel flow is cut off (or substantially reduced) during which time the turbine blades cool off. The burn phase is not long enough to allow the turbine blades to "soak" to the temperature to which they are exposed.

With reference now to the drawings, Fig. 74 shows a gas turbine 510 according to one embodiment of the present invention including a compressor 512 connected to a turbine 514 by a shaft 516, and a combustor 520. Ambient air is fed into the compressor 512 and compressed air is fed therefrom to a surge tank 518 and then to the combustor 520 where it is mixed with fuel from a fuel

BUREAU OMPI WIPO INTERNATIONAL

tank 522 (the tank 522 can be two or more tanks containing two or more different fuels) and burned, and the hot combustion gases are fed to the turbine 514 and then to exhaust. The combustor 520 encloses a combustion chamber including a primary combustion zone 524 and a secondary zone 526. Compressed air from the surge tank 518 is fed to the primary combustion zone 524 by a line or duct 528, and is fed to the secondary zone 526 by a line or duct 530. The fuel is fed from the fuel tank 522 to the primary combustion zone 524 by a line 532. The flow through the ducts 528 and 530 and the line 532 is controlled by flow regulating means such as valves 534, 536 and 538 respectively. Any one of a large number of known flow regulators or valves can be used, as will be evident to those skilled in the art. The valves can all be of the same type or of different types. They can be controlled mechanically or electrically, and they can be interconnected or independent. Regarding the valves 534 and 536 in the ducts 528 and 530, these "valves" can louvers, gates, valves, etc.

In the embodiment shown in Fig. 74, the valves 534, 536, and 538 are controlled together as a group by a throttle 540 through a mechanical linkage. In the embodiment shown in Fig. 74, the air-to-fuel ratio in the primary combustion zone 524 is controlled by the simple mechanical linkage such that the air-to-fuel ratio is maintained substantially constant, even as the fuel rate varies. For example, as the quantity of fuel fed to the zone 524 is decreased by valve 538, a corresponding proportional decrease is made in the quantity of air fed to the primary combustion zone 524 through the valve 534 so as to maintain the same air-to-fuel ratio in the zone 524. The mechanical linkage can also provide a predetermined proportional change for the valve 536 as the fuel rate varies.

BUREAU OMPI WIPO INTERNATIONAL

In one preferred embodiment of the present invention, the valve 536 is controlled independently of the valves 534 and 538, so that the quantity of cooling air fed to the secondary zone 526 to limit the turbine 514 temperature can be varied independently of the combustion process controlled by the other two valves 534 and 538. For example, a temperature sensor 552 is located adjacent the turbine 514 and is electrically connected to a controller 554 which in turn is mechanically (or electrically depending on the type of valve) connected to the valve 536. Thus, the valve 536 can be controlled in response to the sensed temperature adjacent the turbine 514 to accomplish the desired task of limiting the turbine temperature, independently of the combustion process (i.e. independently of the state of the other two valves 534 and 538). In this embodiment, the valve 536 can also be mechanically linked to the throttle 540 and to the other two valves (as shown, in which case the controller 554 includes means for overriding such mechanical linkage), or the valve 536 can be controlled solely by and connected solely to the controller 554.

Fig. 75 is a graph showing various combustion processes of the present invention. Fig. 75A shows the fuel rate for a conventional prior art gas turbine in which the upper horizontal line shows a high throttle position and the lower horizontal line shows a low throttle position. Fig. 75B, on the other hand, shows a high throttle operation for an intermittent burn operation cycle gas turbine. Fig. 75C shows one process for operating an intermediate burn operation cycle gas turbine with a low throttle, while Fig. 75D shows a different process for operating an intermittent burn operation cycle gas turbine with a low throttle. According to the present invention, in Figs. 75B, 75C and 75D, the air-to-fuel

BUREAU OMPI WIPO INTERNATIONAL

ratio is the same in all cases. In Figs. 75B and 75C the fuel rate is the same, while in Fig. 75D the fuel rate is substantially less than in Figs. 75B and 75C, however, the length of the burn phase in Fig. 75D is substantially greater than it is in Fig. 75C. It is also to be noted that the burn temperature is maintained substantially constant and is substantially the same for each of the operations shown in Figs. 75B, 75C and 75D.

Fig. 76 shows another combustion process embodiment of the present invention. Fig. 76A shows a gradually depressed accelerator pedal and Fig. 76B shows just one of the many different possible combustion process responses according to the present invention. (in the standard prior art conventional gas turbine engine, the response would be a gradually increased fuel rate with a varying air-to-fuel ratio and a varying burn temperature). Fig. 76B shows a gas turbine operated according to the present invention with an intermittent burn operation cycle in which the air-to-fuel ratio is maintained constant as is the burn temperature. In this particular embodiment, the length of time of the burn phase and the length of time of the cool phase is also maintained constant. The only change is in the quantity of fuel fed to the primary combustion zone (the fuel rate).

Fig. 77 shows another response to the gradually depressed accelerator pedal shown in Fig. 77A. Fig. 77B shows a response in which in the first two burn phases everything is maintained the same, followed by three burn phases in which the length of the burn phase is shortened, the length of the cool phase remains the same, the burn temperature and the air-to-fuel ratio remains the same, but the fuel rate is increased. The next two burn phases show again an increase in the fuel rate,

BUREAU OMPI WIPO INTERNATIO

but substantially the same length of burn phase and cool phase. The last two burn phases show a greatly increased fuel rate with substantially the same length of burn phase and cool phase.

A comparison of Figs. 3B and 4B illustrate the great flexibility of responses to a particular throttle condition, in this case a gradually increased throttle demand.

Fig. 78 shows different combustion processes of the present invention with Fig. 78A showing one fuel option, Fig. 78B showing one air option, Fig. 78C showing a different air option and Fig. 78D showing a still different fuel option, all for the same throttle demand. It should be noted, between Figs. 78A and 78D, for example, that for a given throttle demand the fuel graphs can look substantially different. Fig. 78A shows a lower fuel rate but a longer burn phase, while Fig. 78D shows a larger fuel rate, a shorter burn phase and a longer cool phase. In both Figs. 78A and 78D, the air-to-fuel ratio and the burn temperature can be the same. Comparing Figs. 78B and 78C, in Fig. 78B, the amount of air fed to the combustion chamber is substantially constant, whereas in Fig. 78C the quantity of air fed to the combustion chamber has two different levels; a first lesser quantity during the burn phase, and second greater quantity during the cool phase. This increase in air during the cool phase can be accomplished solely by the valve 536 in Fig. 74 or solely by the valve 534, or it can be by a combination of both valves. However, when an increased quantity of cooling air is desired during the burn phase, it can be provided, however, it would be provided solely by the valve 536 because, at a given fuel rate, the valve 534 cannot be changed without changing the air-to-fuel ratio in the primary combustion zone.

BUREAU OMPI WIPO INTERNATIONAL

Referring now to Fig. 79, Fig. 79 shows in Fig. 79A one fuel option during an intermittent burn cycle and Figs. 79B and 79C show two different air options that can be used with the Fig. 79A fuel option. In Fig. 79C it will be seen that the air quantity is increased during the cool phase of the intermittent burn cycle.

Figs. 80 and 81 similarly show various air and fuel options according to the present invention.

Fig. 82 is a graph showing a variety of different fuel options for a high throttle condition. It will be noted in Fig. 82D that the fuel rate may even be lower than it is in certain low throttle options, see for example Fig. 78D, however, the distinction is that in Fig. 82D the burn phase is much longer than it is in the low throttle option in Fig. 78D.

Fig. 83 shows a particular intermittent burn operation cycle with Fig. 83A showing the fuel rate of a burn phase followed by a cool phase. Fig. 83B shows the air rate with increased amounts of air during the cool phase. At point 50 in Fig. 83B, the sensor 552 (Fig. 74) has sensed that the temperature of the turbine 514 is lower than a predetermined temperature. The control means 554 then closes the valve 536 by an amount to reduce the amount of cooling air fed to the combustion chamber 520. In other words, Fig. 83 shows an embodiment of the present invention providing an additional degree of flexibility wherein the cycle timing is different for the cooling phase air from the combustion air and combustion fuel cycle. Thus, the cooling can be tied to the sensor 552 for modulating the amount of additional cooling air fed to the combustion chamber 520 during the cool phase of the intermittent burn cycle.

It is to be noted that all of the graphs in Figs. 75-83 use straight lines meeting perpendicularly, however, this is only for convenience in describing the invention. These "wave forms" can vary greatly in shape,

BUREAU OMPI WIPO INTERNATIONAL

can be curved and/or saw-toothed in all possible combinations. That is, the fuel rate and the air rate can increase and decrease in a great variety of ways depending on how to best satisfy all requirements.

The invention has been described in detail with particular reference to the preferred embodiments thereof, but it will be understood that variations and modifications can be effected within the spirit and scope of the invention as described hereinabove and as defined in the appended claims.

For example, the present invention is useful for any and all applications for gas turbines: automobiles, locomotives, central station power, jet engines, tanks, trucks, mobile power supplies, etc. The preferred air-to-fuel ratio may vary from one application to another, for example, to meet different requirements such as performance and temperature constraints and pollution requirements (which may vary from year to year and state to state), although the preferred ratio is in the range of about 15:1 to about 35:1. This invention allows the burn temperature to be maximized and maintained at the maximum value, even as the fuel rate varies. The maximum value is approximately the value at a stoichiometric air-to-fuel mixture ratio. The present invention does not replace the pulse width modulation techniques of applicant's said copending application Serial No. 879,969, but rather supplements the previous techniques. All throttle modulation methods have their place: vary fuel rate, length of burn phase, length of cool phase, and amount of cooling air. It is important to note that the maximum fuel rate need not be used during burn to achieve the maximum possible gas temperature, for maximizing efficiency. While a simply mechanical linkage is shown to provide one substantially constant air-to-fuel ratio in the primary combustion zone and another ratio in the secondary zone, simple linkage from the throttle

BUREAU OMPI WIPO INTERNATIONAL

pedal to appropriate sensors like combustion temperature, turbine temperature, etc. can be used (analogous, for example, to today's automatic choke). More sophisticated systems can be used, as will be understood by those skilled in the art, as demanded by the application. For example, central station power generation may dictate the use of a microprocessor. Louvers, gates, valves, etc. can be used with the surge tank 518 as desired. In Fig. 74 all of the compressed air is fed to the surge tank, however, this is not essential and other configurations with suitable valving can be used. It is noted that the gasifier section shown in Fig. 74 at a 5:1 pressure ratio, would be under about 75 psig. The air-to-fuel ratio maintained in the primary combustion zone 524 can be varied during operation in response, for example, to changes in ambient temperature, changes in type of fuel used, and changes in type of driving (such as from continuous high speed to stop and go), by use of suitable sensors and controls such as ambient temperature sensor. The "direct" and immediate control of the air-to-fuel ratio is in contrast to the indirect and time lagging indirect change (at no particular ratio) in the prior art. The "burn" phase of the intermittent burn cycle is in sharp contrast to the prior art gas turbine that used explosions of fuel, usually in a constant volume. There was no continuous burning at substantially constant pressure as in the intermittent burn-cool cycle of the present invention and of that described in applicant's said copending application Serial No. 879,969. It is noted that additional air fed to the combustion chamber from the surge tank 518 during the cool phase of the intermittent cycle can supply part or all of the flywheel force to the gas turbine.

BUREAU OMPI WIPO INTERNATION

Further, while the above description shows the air fed to the primary combustion zone coming from the compressor (via the surge tank), this amount of air may be much less than that fed to the secondary zone. In this case, the primary combustion zone need not be at the high pressure of the surge tank and of the secondary air; the primary air can be at atmospheric pressure (or a slight pressure greater than atmospheric by using a small blower). According to this embodiment, the burn chamber will be a relative low pressure and can be separate from the secondary zone through which the greater volume of much higher pressure secondary air flows. The two chambers can be connected and the hot combustion gases can then flow into the secondary zone at least partly by an aspirating action.

When a heat exchanger or regenerator (for using the exhaust heat) is desired, it can be used with any one (or more) of the air duct 530, the air duct 528, or the fuel line 532.

The term "gas motor" herein means a motor that operates on gas and has a stator, a rotor connected to a shaft, a gas inlet and a gas outlet, such as: (1) a rotary sliding vane device, which is a positive displacement motor, and (2) a dynamic turbine as used in gas turbines, which is not a positive displacement device.

The invention has been described in detail with particular reference to the preferred embodiments thereof, but it will be understood that variations and modifications can be effected within the spirit and scope of the invention as described hereinabove and as defined in the appended claims.

BUREAU OMPI WIPO INTERNATIONAL

What is claimed is:

1. A positive displacement Brayton cycle engine comprising:

(a) a source of compressed gas;

(b) a constant pressure combustion chamber including a gas inlet, a gas outlet and means for burning fuel therein for heating gas fed into said combustion chamber;

(c) a gas motor having a gas inlet and a gas outlet and connected to a rotatable shaft;

(d) at least one of said source and said motor including a rotary sliding vane device; and

(e) means for operating said engine in a substantially constant pressure Brayton cycle including means for feeding a continuous flow of gas from said source to said combustion chamber, means for continuously heating said gas in said combustion chamber at substantially constant pressure, means for feeding a continuous flow of said hot gas from said combustion chamber solely to said motor, and means for continuously exhausting gas from said motor.

2. The apparatus according to claim 1 wherein said operating means includes means for operating said engine in an open cycle and wherein said gas is air and said exhausting means includes means for exhausting said gas to atmosphere.

3. The apparatus according to claim 1 wherein said source includes a constant output pressure rotary sliding vane compressor.

4. The apparatus according to claim 3 wherein said source also includes a surge tank connected to said compressor and means for feeding gas from said surge tank to said combustion chamber.

BUREAU OMPI WIPO INTERNATIONAL

5. The apparatus according to claim 4 including means for heat exchanging the heat in the exhaust gas with the gas being fed into said combustion chamber.

6. The apparatus according to claim 5 wherein said operating means includes means for operating said engine in an open cycle and wherein said gas is air and said exhausting means includes means for exhausting said gas to atmosphere.

7. The apparatus according to claim 3 wherein said motor is a dynamic turbine.

8. The apparatus according to claim 1 wherein said motor includes a rotary sliding vane motor.

9. The apparatus according to claim 8 wherein said source includes a constant output pressure rotary sliding vane compressor.

10. The apparatus according to claim 9 wherein said motor and compressor are connected to the same shaft, whereby rotation of said motor causes said compressor to rotate.

11. The apparatus according to claim 8 wherein said rotary sliding vane motor includes means for providing continuity of gas flow therethrough.

12. The apparatus according to claim 8 wherein said rotary sliding vane motor includes means for maintaining the gas flowing therethrough in fluid communication with a motor inlet or a motor outlet.

13. The apparatus according to claim 8 wherein said motor has at least one sliding vane and wherein said at least one sliding vane is made out of graphite.

BUREAU OMPI WIPO INTERNATIONAL

14. The apparatus according to claim 1 wherein said source includes a rotary sliding vane compressor having a gas inlet for gas to be compressed, a compressed gas outlet and a rotatable shaft, said compressor including means for maintaining the gas flowing therethrough in fluid communication with at least one of said inlet and said outlet, and wherein said compressor includes a stator defining a chamber therein, a rotor having at least one sliding vane thereon and being mounted for rotation on said shaft and located in said chamber, said stator and rotor defining therebetween at least one compression chamber having a decreasing volume in the direction of rotation of said rotor, said gas inlet being a gas inlet in said stator in fluid communication with a large volume end of said compression chamber, and said gas outlet being a gas outlet in said stator in fluid communication with a small volume end of said compression chamber, and wherein said maintaining means comprises said rotor having a maximum of one sliding vane between said compressor inlet and said compressor outlet at all rotational positions of said rotor.

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15. The apparatus acccrding to claim 14 wherein said compression chamber continuously decreases in volume along the entire length thereof from said compressor inlet to said compressor outlet.

16. The apparatus according to claim 1 wherein said operating means comprises means for operating said engine in a closed cycle and wherein said exhaust gas feeding means includes means for feeding said exhaust gas back to said source, and wherein said combustion chamber is an indirect combustion chamber.

17. The apparatus according to claim 16 including means for heat exchanging the exhaust gas with the gas fed from said source to said combustion chamber outlet.

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18. The apparatus according to claim 1 wherein said source includes a rotary sliding vane compressor connected to a surge tank for feeding compressed air to said surge tank including means for feeding gas from said surge tank to said combustion chamber, wherein said compressor has at least one sliding vane, and wherein said at least sliding vane is made out of graphite.

19. The apparatus according to claim 1 including means for generating a hot-cool operating cycle comprising a hot phase continuously alternating with a cool phase, the gas fed to said motor during said hot phase being hot and the gas fed to said motor during said cool phase being cooler.

20. The apparatus according to claim 19 wherein said generating means includes means for continuously feeding gas to said combustion chamber during both said hot phase and said cool phase.

21. The apparatus according to claim 19 including means for maintaining the speed of said shaft during said cool phase substantially the same as that of said shaft during said hot phase, for providing uniform and continuous output.

22. The apparatus according to claim 19 including means for varying the length of at least one of said hot and cool phases in response to throttle changes.

23. The apparatus according to claim 1 including means for maintaining the temperature of the gas flowing through said motor inlet as the highest gas temperature in said engine.

24. The apparatus according to claim 1 including means for maintaining the burn temperature in said combustion chamber substantially constant even as the rate fuel is fed to said combustion chamber varies.

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25. The apparatus according to claim 1 including means for feeding a continuous flow of gas to said source, means for feeding a continuous flow of gas to said combustion chamber, and means for feeding a continuous flow of gas from said combustion chamber to said motor.

26. The apparatus according to claim 1 wherein said source is a positive displacement, rotary sliding vane compressor having an inlet and an outlet and connected to said rotatable shaft, wherein said motor is a positive displacement, rotary sliding vane motor having an inlet and an outlet, and wherein said compressor and said motor comprise a single, positive displacement, rotary sliding vane compressor-motor comprising a stator defining a chamber therein, a rotar in said housing having at least one sliding vane thereon and being mounted for rotation on said shaft and located in said chamber, said stator and rotor defining therebetween at least one compressor portion having a decreasing volume chamber in the direction of rotation of said rotor and at least one motor portion having an increasing volume chamber in the direction of rotation of said rotor.

27. The apparatus according to claim 26 including:

said stator having a first port in fluid communication with a large volume portion of said decreasing volume chamber for feeding gas thereinto whereby it will be compressed, said first port being said gas inlet;

said stator having a second port in fluid communication with a small volume portion of said decreasing volume chamber for feeding the compressed gas out of said decreasing volume chamber, said second port being said compressor gas outlet;

BUREAU OMPI WIPO INTERNATIONAL

said stator having a third port in fluid communication with a small volume portion of said increasing volume chamber for feeding a hot gas thereinto for causing rotation of said rotor, said third port being said motor inlet;

said stator having a fourth port in fluid communication with a large volume portion of said increasing volume chamber for exhausting hot gas therefrom, said fourth port being said motor gas outlet.

28. The apparatus according to claim 27 wherein said at least one sliding vane is made of graphite.

29. The apparatus according to claim 27 including a compression spring for biasing each of said at least sliding vane in a radially outward direction.

30. The apparatus according to claim 27 wherein said engine is an open cycle engine and including means for feeding the exhaust gas from said fourth port to atmosphere.

31. The apparatus according to claim 27 wherein said engine is a closed cycle engine and including means for feeding the exhaust gas from said fourth port back to said first port and including means for heat exchanging said exhaust gas with the gas fed from said second port to said combustion chamber and wherein said combustion chamber is an indirect combustion chamber.

32. The apparatus according to claim 27 including means for feeding fuel into said first port.

33. The apparatus according to claim 27 including means for heat exchanging the heat from the hot gas introduced into said third port to the compressed gas fed out said second port.

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34. The apparatus according to claim 33 wherein said third port is closer to said second port than to said first port to provide improved heat exchange between the hot gas fed into said third port and the compressed gas fed out of said second port.

35. The apparatus according to claim 27 including heat radiating fins on said stator on at least the outside surface thereof adjacent said motor portion of said compressor-motor.

36. The apparatus according to claim 27 including means for regenerating a portion of the gas fed out of said second port back into said first port.

37. The apparatus according to claim 26 including means for balancing the motor-compressor gas ratio.

38. The apparatus according to claim 37 including a surge tank connected to said second port, whereby compressed gas is fed into said tank.

39. The apparatus according to claim 37 wherein said stator has an oval shape and wherein said motor portion occupies a larger percentage of said chamber in said stator than does said compressor portion.

40. The apparatus according to claim 26 wherein said rotor is tangent to said stator at two locations defining two separate working chambers between said rotor and stator.

41. The apparatus according to claim 40 wherein one of said working chambers is a compressor chamber and the other is a motor chamber.

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42. The apparatus according to claim 41 wherein the compressor inlet opens into about half of said compressor chamber, whereby said compressor chamber is of continuously decreasing volume.

43. The apparatus according to claim 41 wherein said motor outlet opens into about half of said motor chamber, whereby said motor chamber is of continuously increasing volume.

44. The apparatus according to claim 41 wherein said rotor includes four sliding vanes.

45. The apparatus according to claim 44 wherein said vanes are equally spaced-apart.

46. The apparatus according to claim 41 wherein the volume of said motor chamber is greater than the volume of said compressor chamber.

47. The apparatus according to claim 40 including means for generating a hot-cool operating cycle comprising a hot phase continuously alternating with a cool phase, the gas fed to said motor during said hot phase being hot and the gas fed to said motor during said cool phase being cooler.

48. The apparatus according to claim 47 including means for varying the length of at least one of said hot and cool phases in response to throttle changes.

49. The apparatus according to any one of claims 1, 9, 11, 12, 19, or 26 wherein said motor includes a stator defining a chamber therein, a rotor having at least one sliding vane thereon and being mounted for rotation on said shaft and located in said chamber, said

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stator and rotor defining therebetween at least one motor chamber having an increasing volume in the direction of rotation of said rotor, said motor inlet being a gas inlet in said stator in fluid communication with a small volume end of said motor chamber, and said motor outlet being a gas outlet in said stator in fluid communication with a large volume end of said motor chamber, and wherein said rotor has a maximum of one sliding vane between said inlet and outlet at all rotational positions of said rotor.

50. The apparatus according to any one of claims 3, 8, or 9 including means for generating a hot-cool operating cycle comprising a hot phase continuously alternating with a cool phase, the gas fed to said motor during said hot phase being hot and the gas fed to said motor during said cool phase being cooler.

51. The apparatus according to claim 50 including means for varying the length of at least one of said hot and cool phases in response to throttle changes.

52. A unitary compressor-motor comprising a single, unitary, positive displacement, rotary sliding vane compressor-motor comprising a stator defining a chamber therein, a rotor in said housing having at least one sliding vane thereon and being mounted for rotation on a shaft and located in said chamber, said stator and rotor defining therebetween at least one compressor portion having a decreasing volume chamber in the direction of rotation of said rotor and at least one motor portion having an increasing volume chamber in the direction of rotation of said rotor, said compressor-motor having a compressor inlet for gas to be compressed, a compressor outlet, a motor inlet and a motor outlet and means for maintaining the gas in said compressor and motor in fluid communication with at least one of said respective inlets and outlets.

BUREAU OMPI WIPO INTERNATIONAL

53. The apparatus according to claim 52 wherein said maintaining means comprises said rotor having a maximum of one sliding vane between each of said inlets and outlets at all rotational positions of said rotor.

54. A Brayton cycle gas engine method comprising the steps of:

(a) compressing a gas in a pressurized gas source;

(b) feeding compressed gas from said source into a combustion chamber;

(c) burning fuel in said combustion chamber and heating said compressed gas while maintaining said compressed gas therein at a constant pressure;

(d) feeding hot compressed gas from said combustion chamber to a gas motor including a stator and a rotor connected to a rotatable shaft to cause rotation of said rotor and said shaft;

(e) providing at least one of said source and said motor as a constant displacement rotary sliding vane device;

(f) operating said gas engine in a substantially constant pressure Brayton cycle, including feeding a continuous flow of gas from said source to said combustion chamber, continuously heating said gas in said combustion chamber at a substantially constant pressure, feeding a continuous flow of said hot, compressed gas at a substantially constant pressure from said combustion chamber solely to said motor, and continuously exhausting gas from said motor.

BUREAU OMPI WIPO INTERNATIONAL

55. The method according to claim 54 wherein said operating step comprises operating said engine in an open cycle and wherein said compressing step comprises compressing air.

56. The method according to claim 54 wherein said compressing step comprises compressing gas in a positive displacement, constant outlet pressure, rotary sliding vane compressor.

57. The method according to claim 56 wherein said compressing step including feeding compressed gas from said compressor to a surge tank and feeding compressed gas from said surge tank to said combustion chamber.

58. The method according to claim 57 including heat exchanging the heat in the motor exhaust gas with the gas fed to the combustion chamber.

59. The method according to claim 58 wherein said operating step comprises operating said engine in an open cycle and wherein said compressing step comprises compressing air.

60. The method according to claim 56 wherein said step of feeding hot, compressed gas to a motor comprises feeding said gas to a dynamic turbine gas motor.

61. The method according to claim 56 including the step of feeding fuel into said compressor inlet.

62. The method according to claim 54 wherein the step of feeding gas to a motor comprises feeding gas to a positive displacement, constant inlet pressure, rotary sliding vane motor having an inlet, an outlet and a shaft.

63. The method according to claim 62 wherein said compressing step comprises compressing gas in a positive displacement, constant outlet pressure, rotary sliding vane compressor.

BUREAU OMPI WIPO INTERNATIONAL

64. The method according to claim 63 including connecting both of said motor and said compressor to the same shaft whereby rotation of said motor causes rotation of said compressor.

65. The method according to claim 62 including providing a continuity of gas flow through said rotary sliding vane motor.

66. The method according to claim 62 including maintaining the gas flowing through said motor in fluid communication with one of said inlet or outlet at all times.

67. The method according to claim 66 wherein said motor includes a stator and a rotor mounted in said stator and having at least one sliding vane thereon and wherein said maintaining stop comprises positioning a maximum of one sliding vane between said motor inlet and said motor outlet at all rotational positions of said rotor.

68. The method according to claim 54 including operating said engine in a closed cycle and including feeding said exhaust gas back into said source.

69. The method according to claim 68 including the step of heat exchanging the exhaust gas with the gas fed to said combustion chamber inlet.

70. The method according to claim 54 including the step of generating a hot-cool operating cycle comprising a hot phase continuously alternating with a cool phase, the gas fed to said motor during said hot phase being hot and the gas fed to said motor during said cool phase being cooler.

BUREAU OMPI WIPO INTERNATIONAL

71. The method according to claim 70 including the step of continuously feeding gas to said combustion chamber during both said hot phase and said cool phase.

72. The method according to claim 70 including the step of maintaining the speed of said shaft during said cool phase substantially the same as that of said shaft during said hot phase to provide uniform and continuous output.

73. The method according to claim 70 including the step of varying the length of at least one of said hot and said cool phases in response to throttle changes.

74. The method according to claim 54 including the step of maintaining the temperature of the gas flowing into said motor inlet as the highest gas temperature in said engine.

75. The method according to claim 54 including burning fuel in said combustion chamber and maintaining the burn temperature in said combustion chamber substantially constant even as the rate fuel is fed to said combustion chamber varies.

76. The method according to claim 54 including feeding a continuous flow of gas to said combustion chamber, and feeding a continuous flow of gas from said combustion chamber to said motor.

77. The method according to claim 54 wherein said source includes a positive displacement rotary sliding vane compressor including a stator and a rotor having at least one sliding vane and being mounted in said stator, and said stator having a gas inlet and a compressed gas outlet.

BUREAU OMPI WIPO INTERNATIONAL

78. The method according to claim 77 wherein the step of feeding gas to a motor comprises feeding gas to a positive displacement, constant inlet pressure, rotary sliding vane motor having an inlet, an outlet and a shaft and including the step of providing both said compressor and said motor as portions of a single, unitary positive displacement rotary compressor-motor including stator, a rotor in said stator connected to an output shaft and defining between said rotor and said stator a compressor portion having a volume increasing chamber.

79. The method according to claim 78 including the step of balancing the compressor-motor volume ratio in said stator.

80. The method according to claim 79 including the step of feeding a portion of the gas fed out said compressor outlet back into said compressor inlet.

81. The method according to claim 77 including the step of feeding the compressed gas from the compressor outlet into a gas surge tank.

82. The method according to claim 54 including the step of heat exchanging heat from the hot gas fed out the exhaust outlet of said motor with the gas fed into said combustion chamber.

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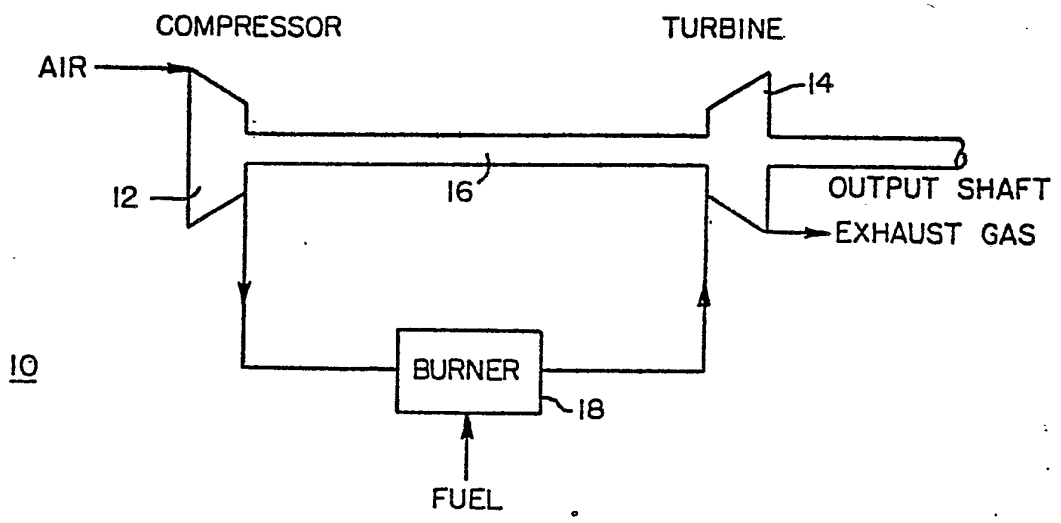


FIG. 1 (PRIOR ART)

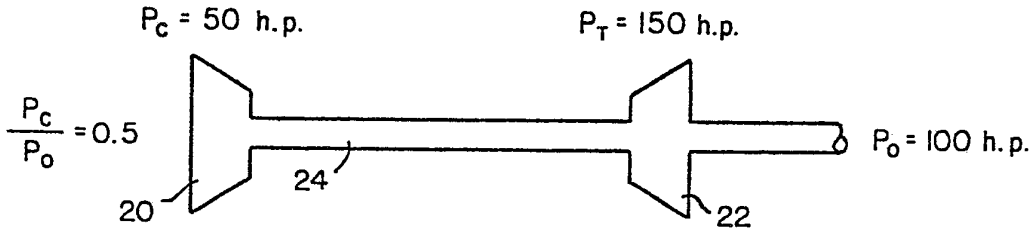


FIG. 2A

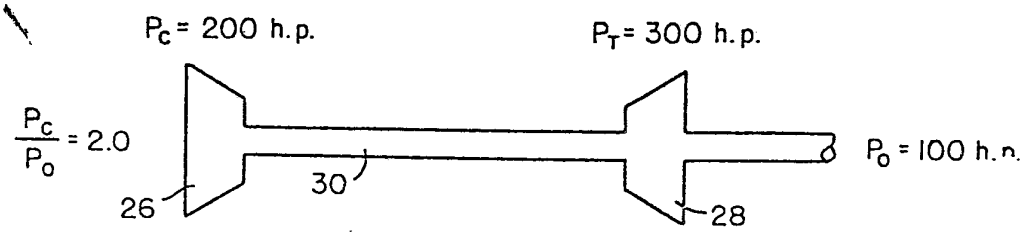


FIG. 2B

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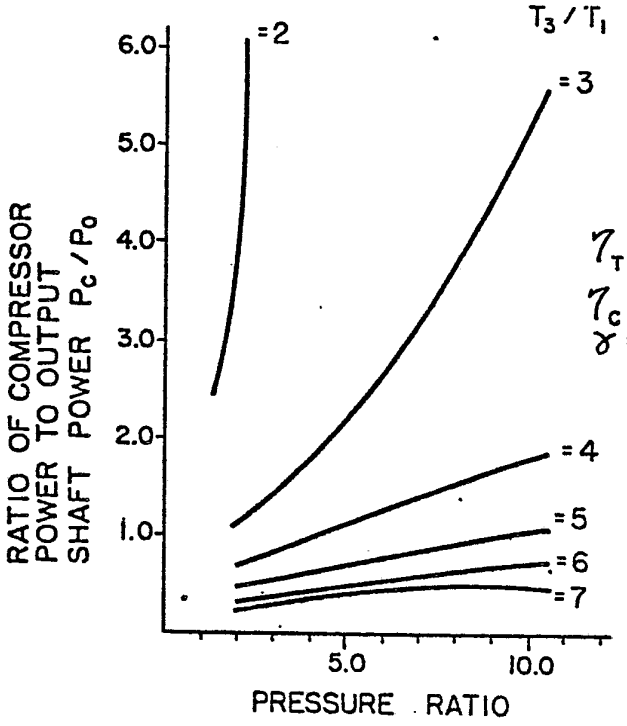


FIG. 3

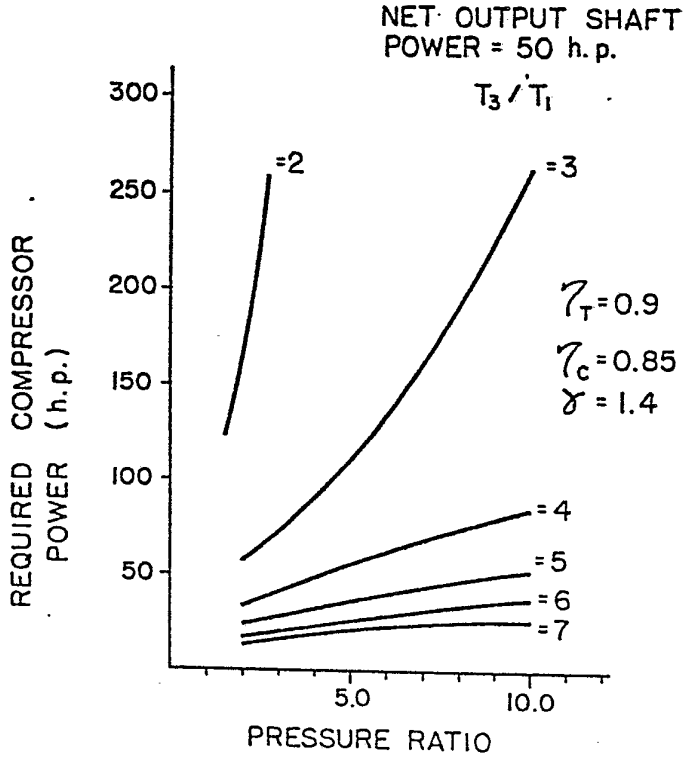


FIG. 4

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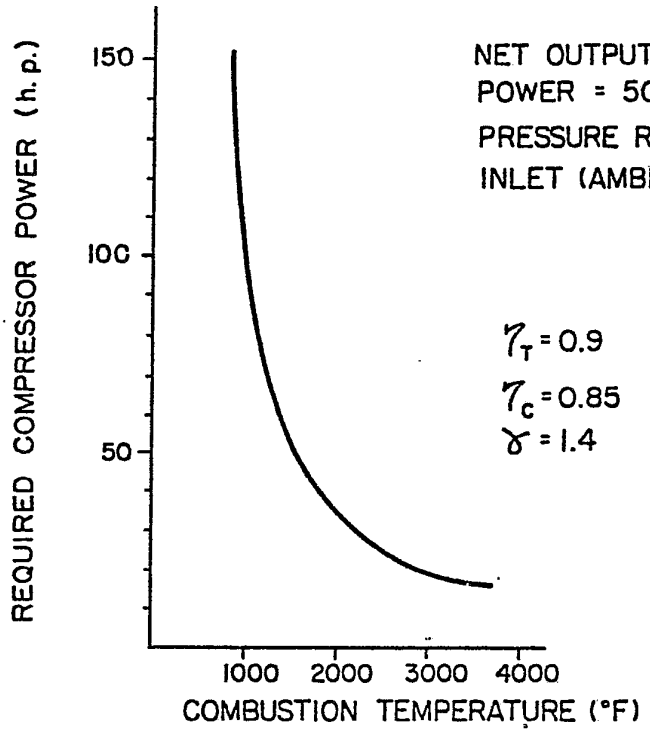


FIG. 5

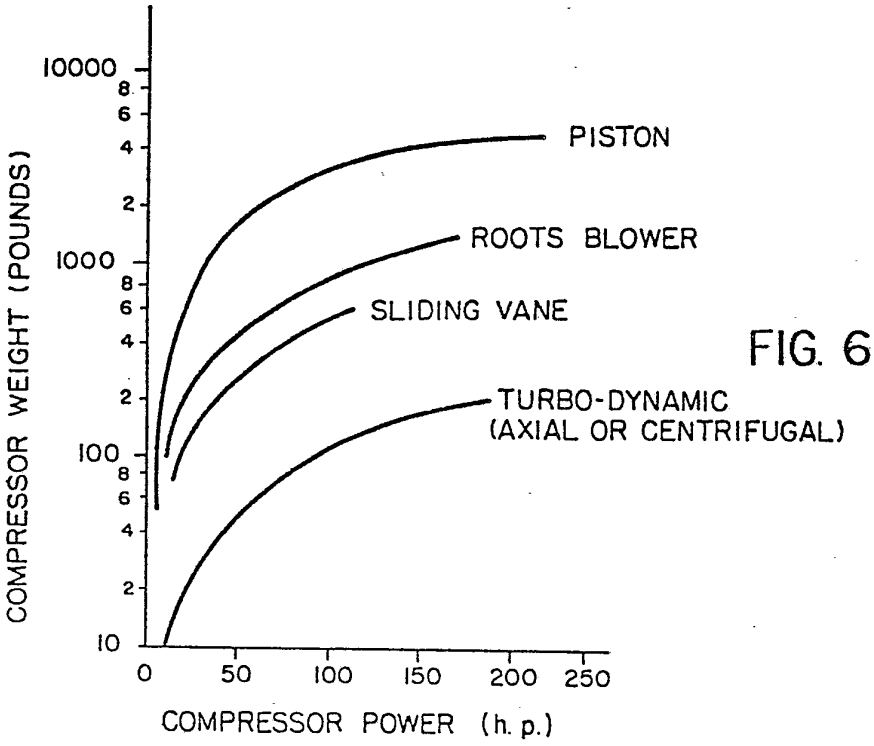


FIG. 6

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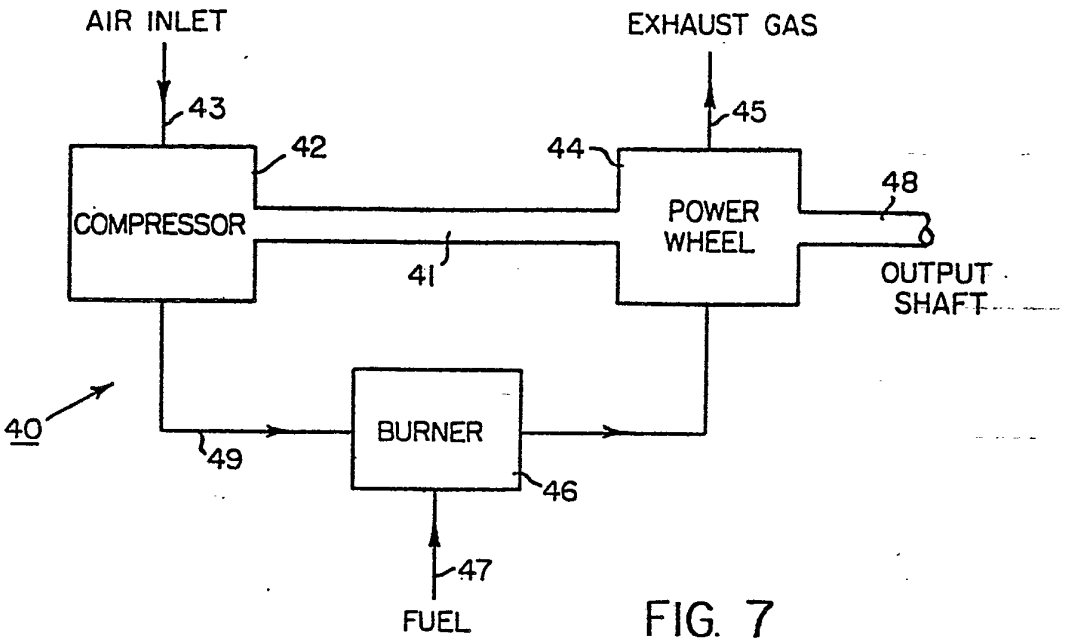


FIG. 7

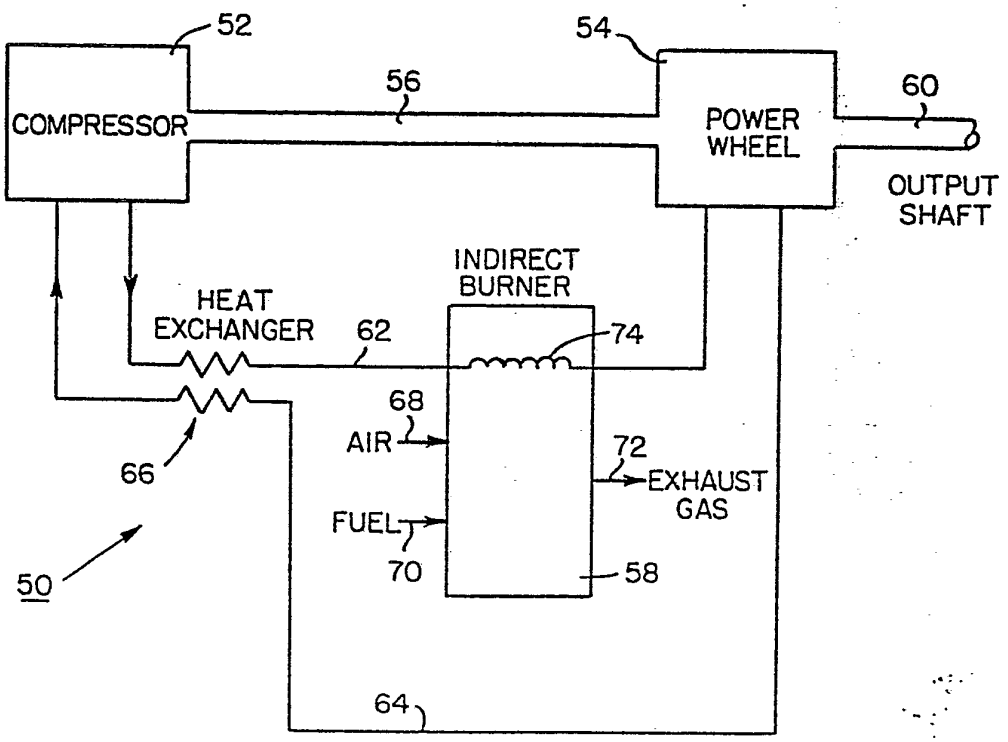


FIG. 8

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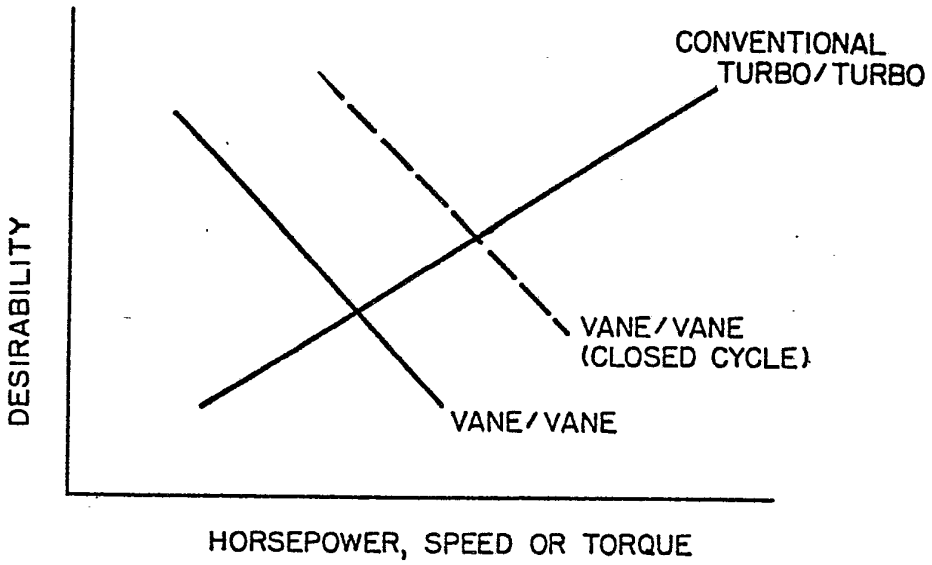


FIG. 9

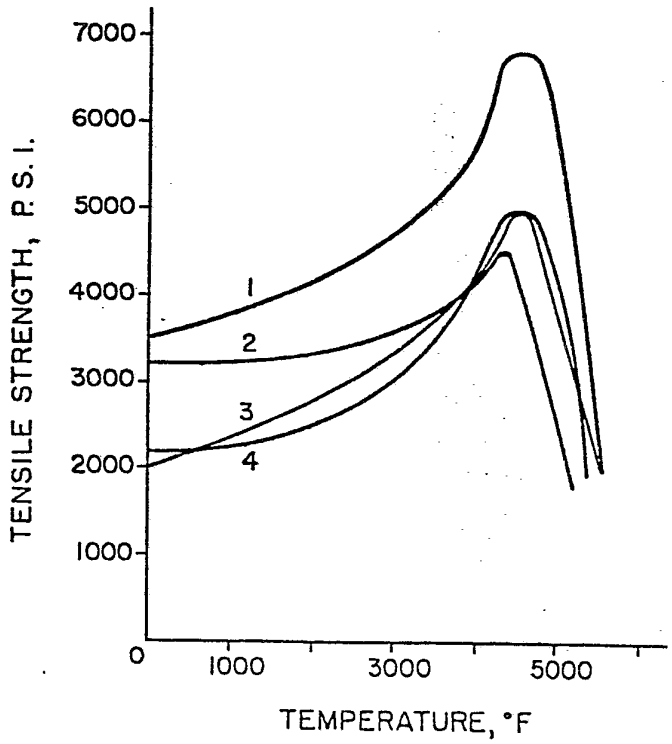


FIG. 10

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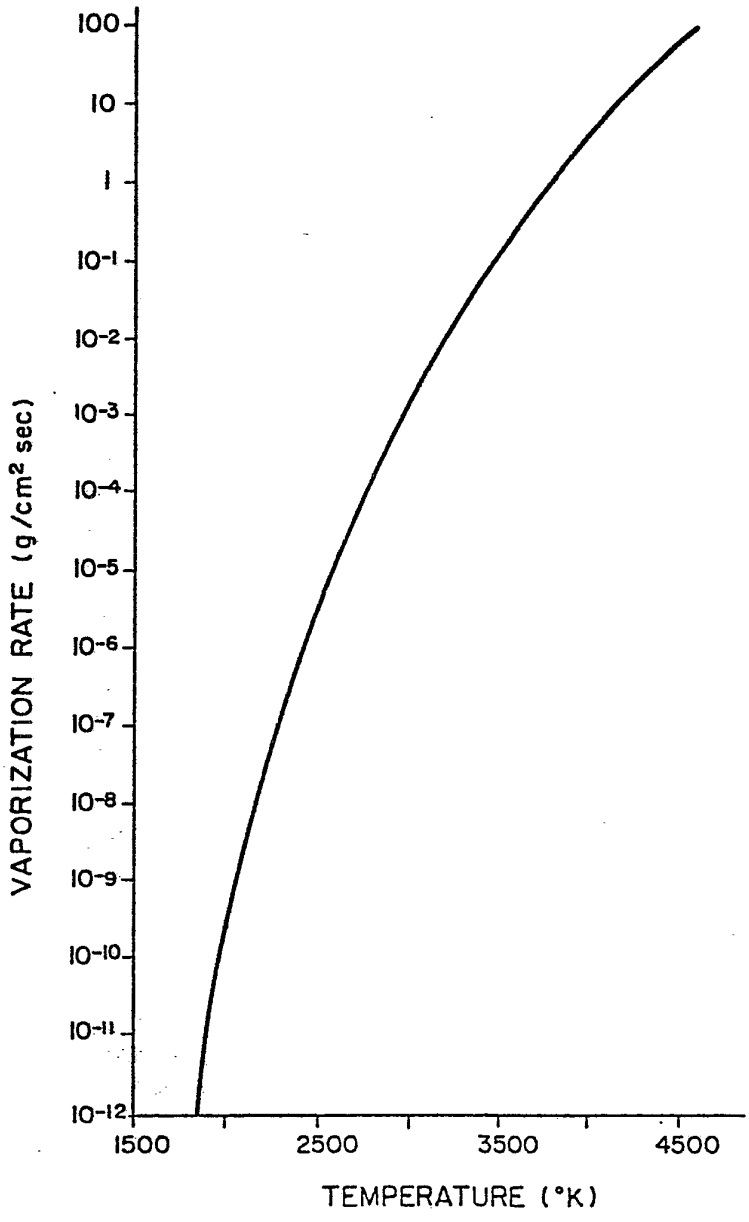


FIG. 11

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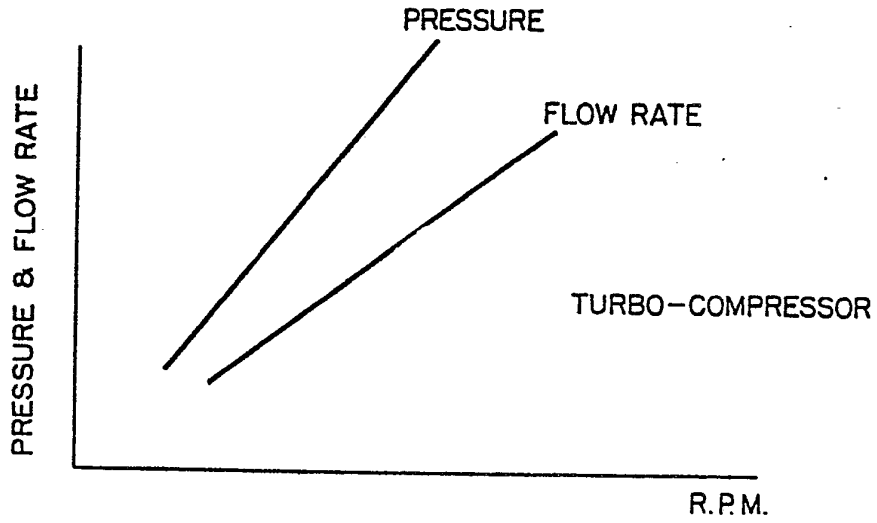


FIG. 12

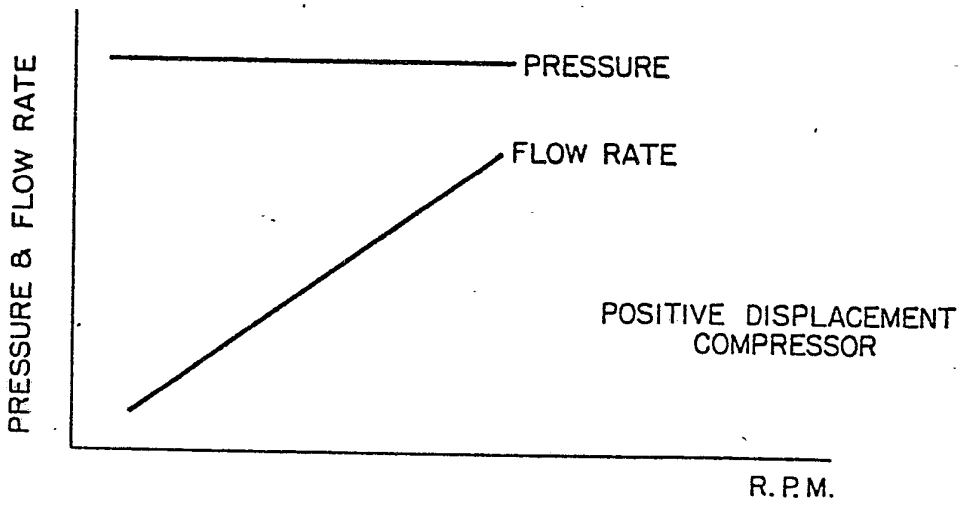


FIG. 13

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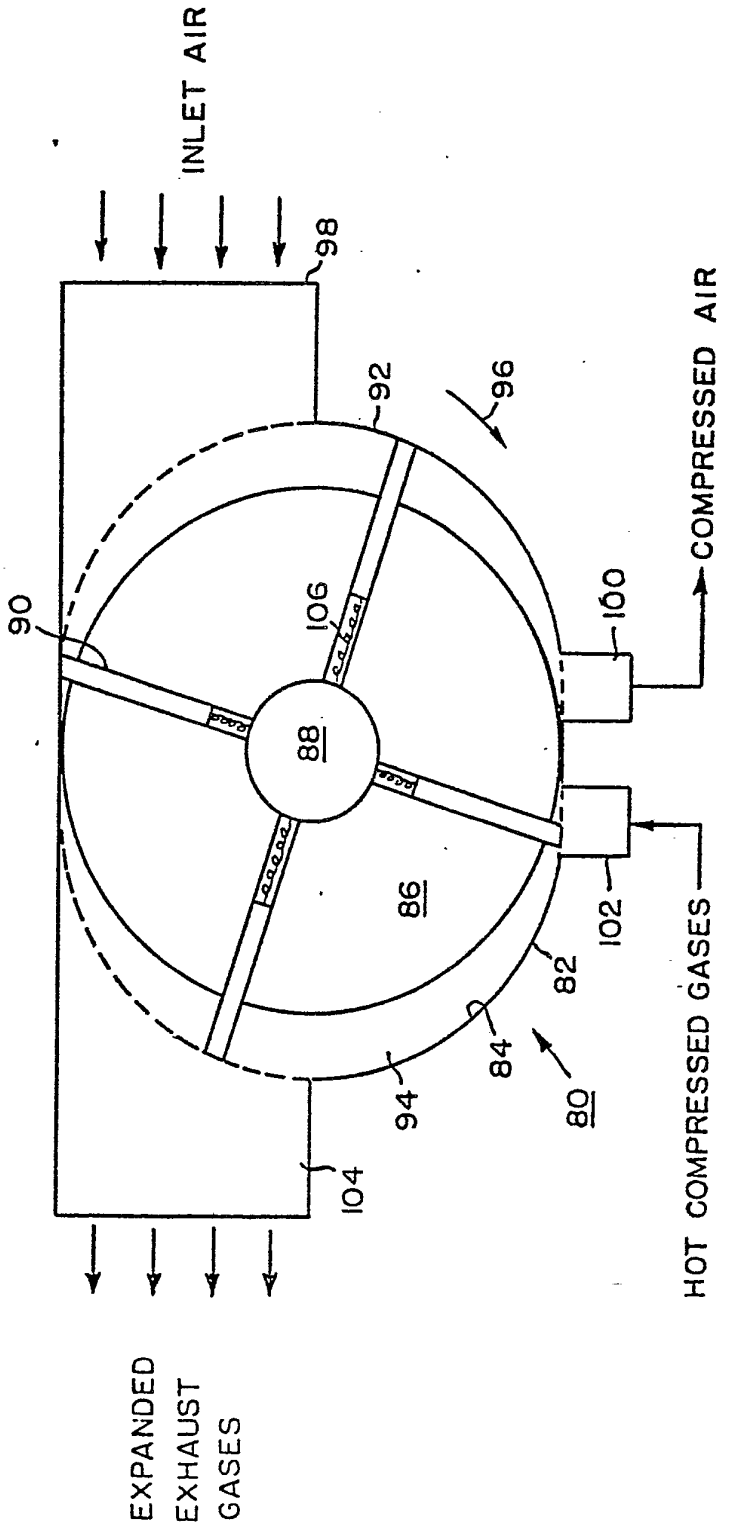


FIG. 14A

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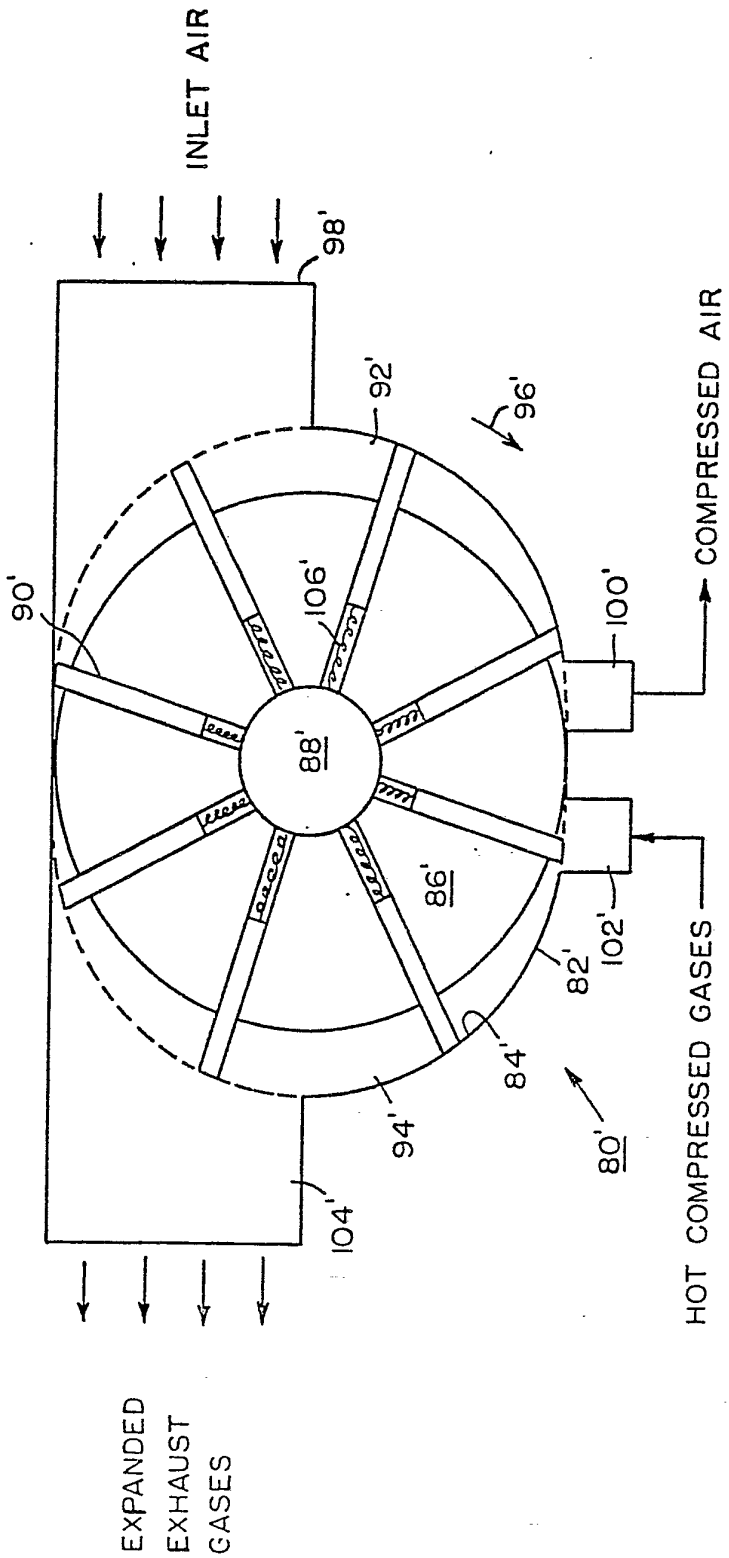


FIG. 14B

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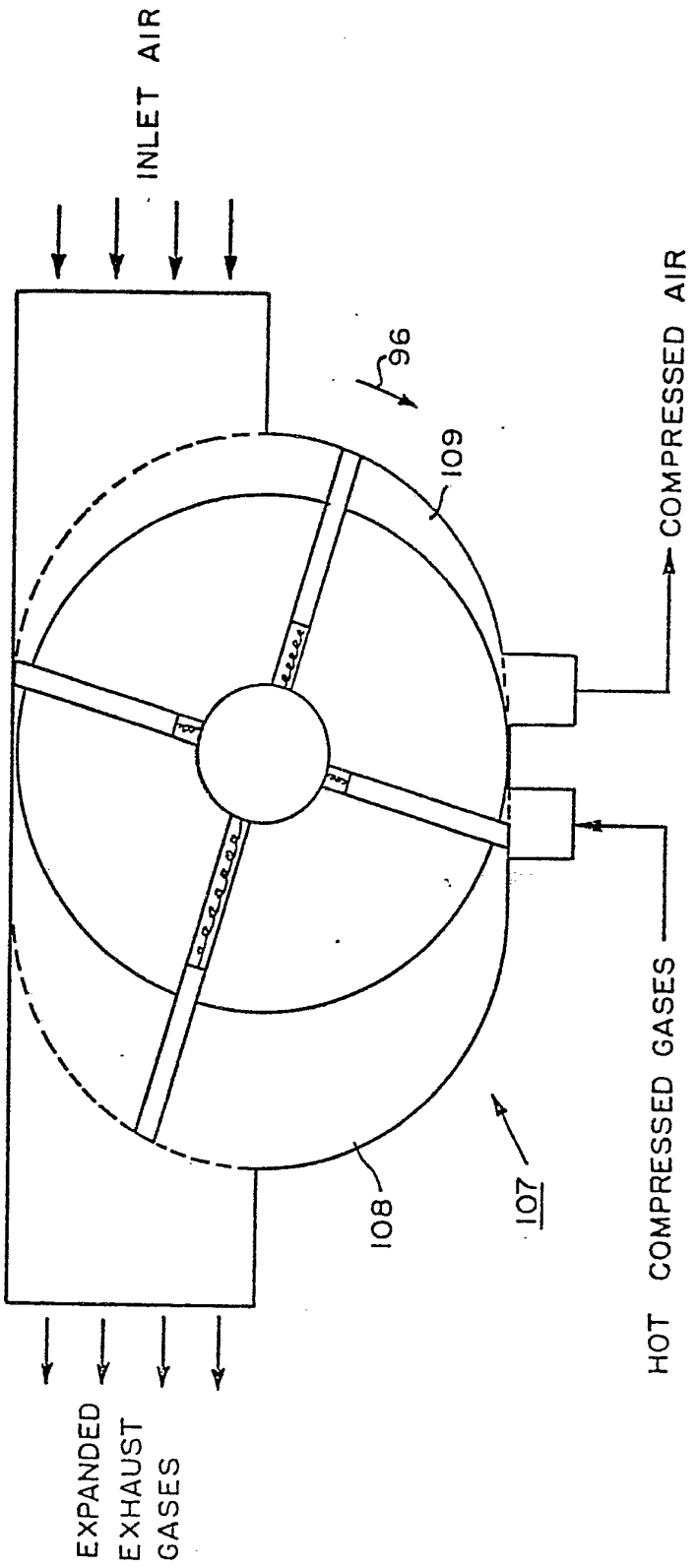


FIG. 15A

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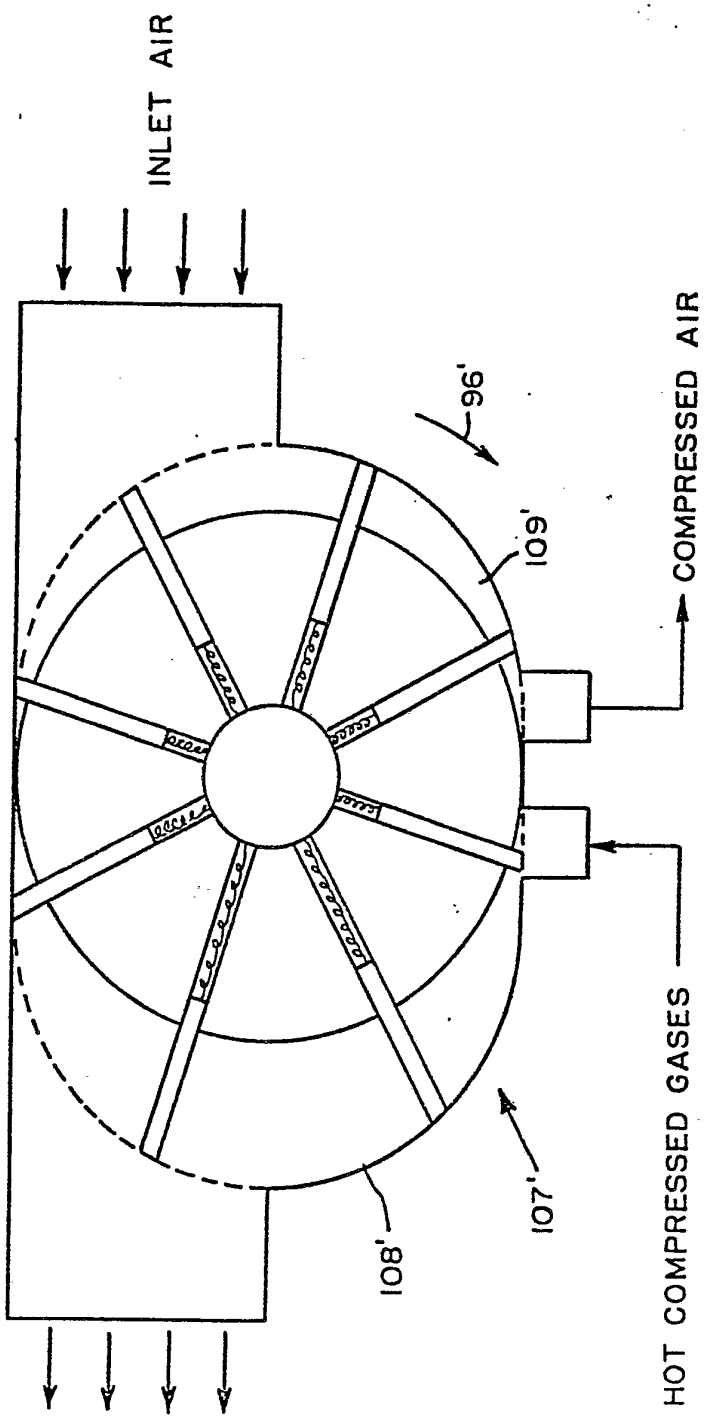


FIG. 15B

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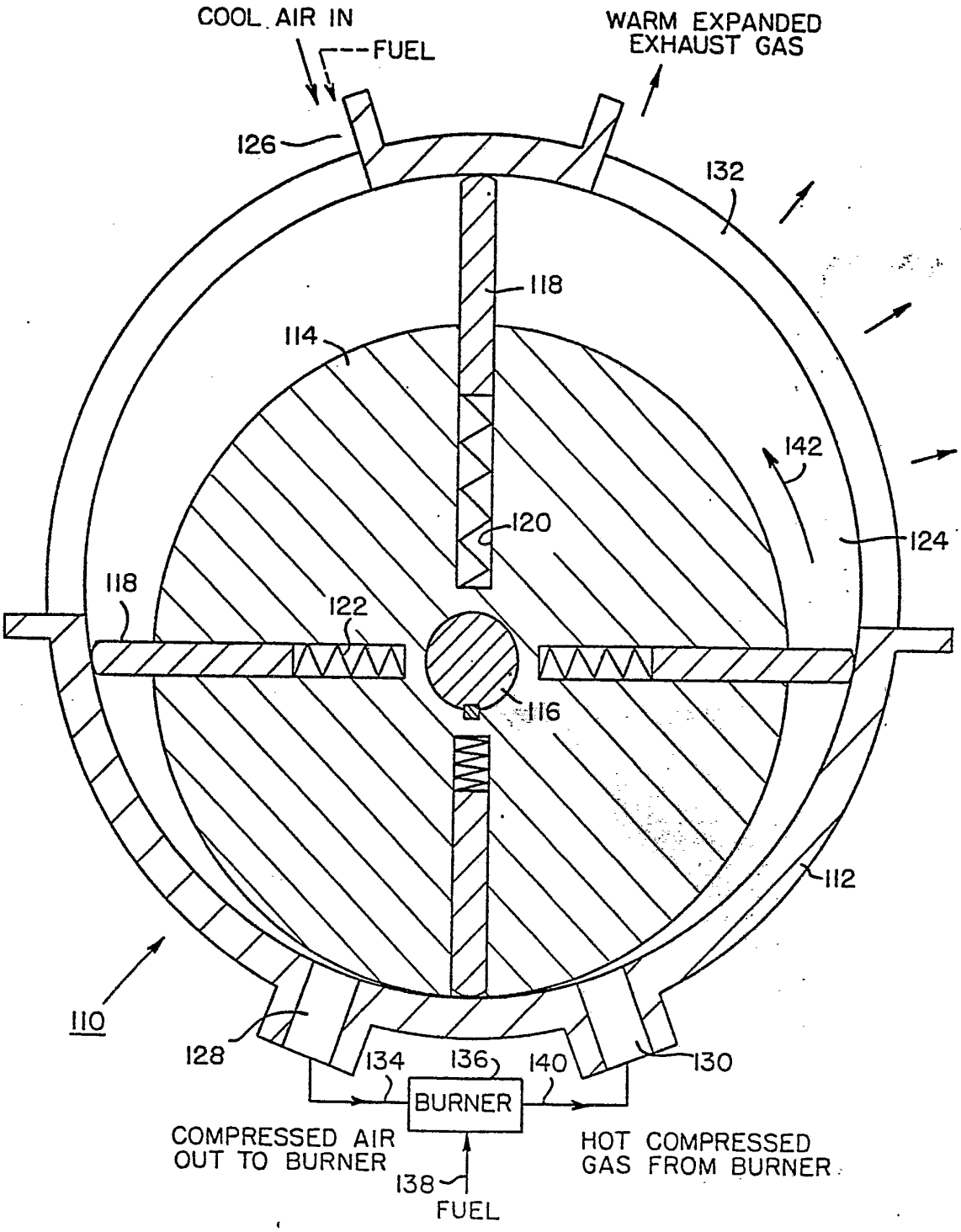


FIG. 16A

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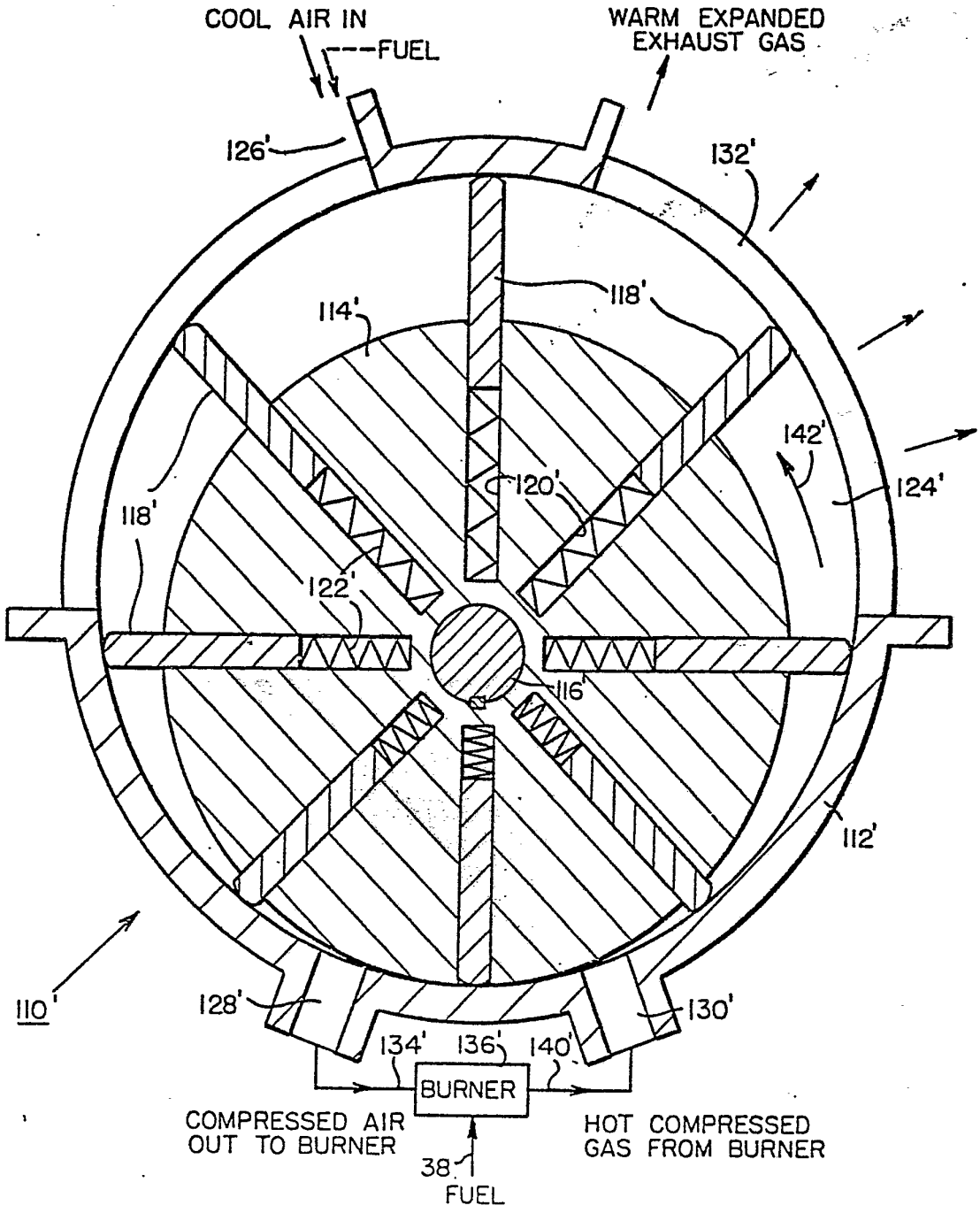
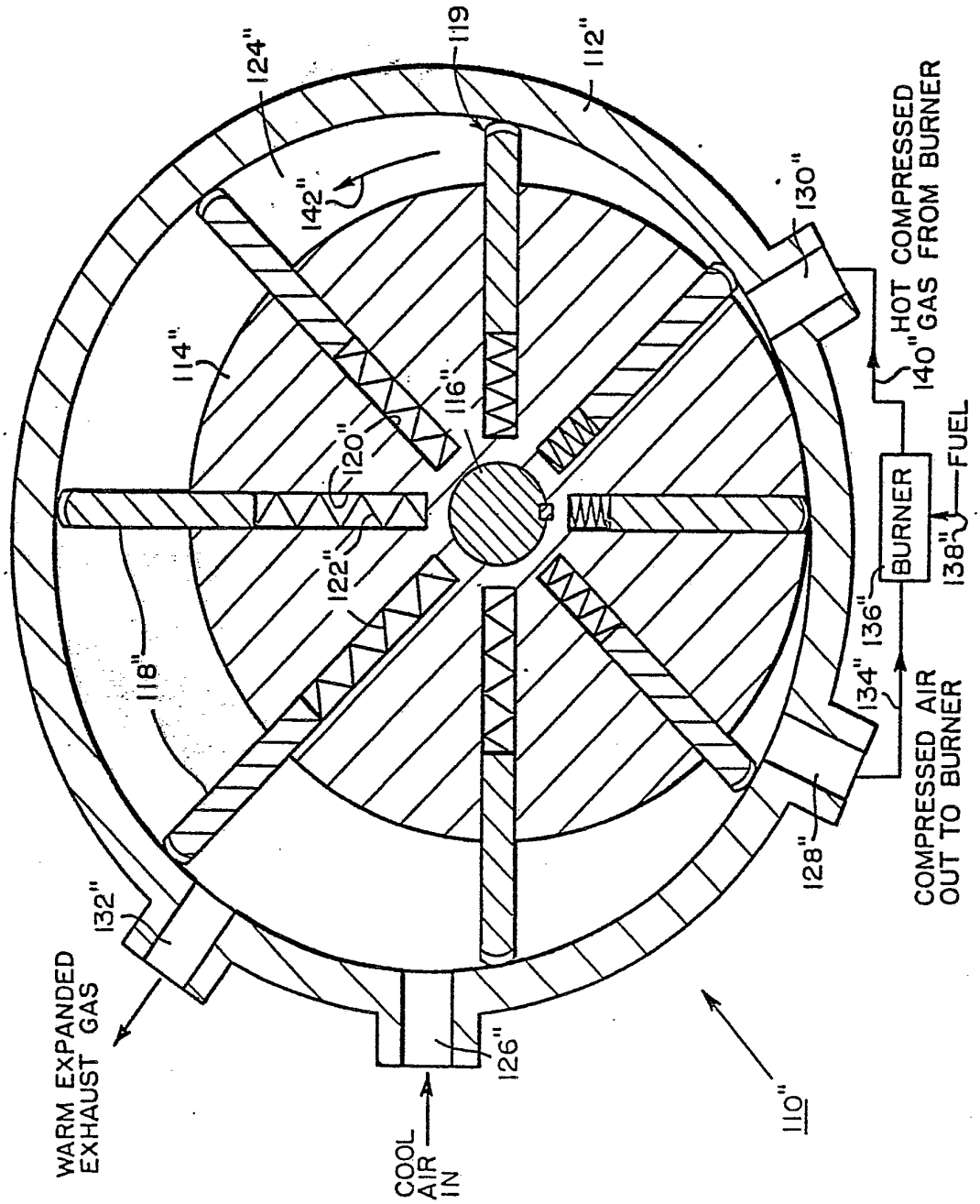


FIG. 16B

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FIG. 17



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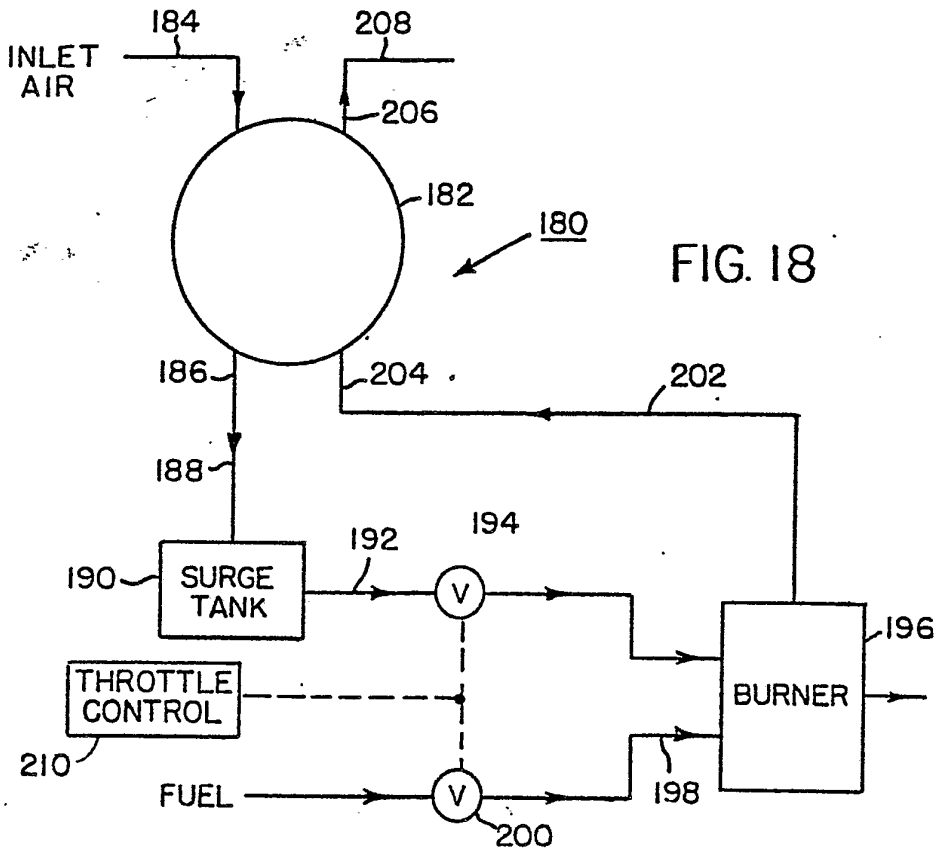


FIG. 18

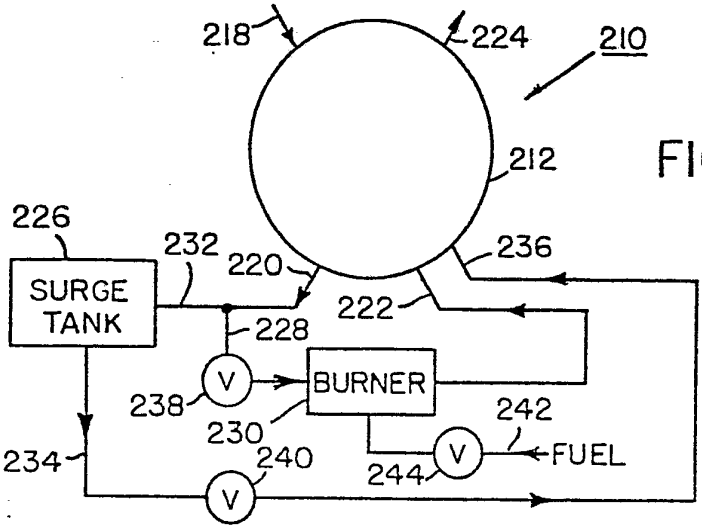


FIG. 19

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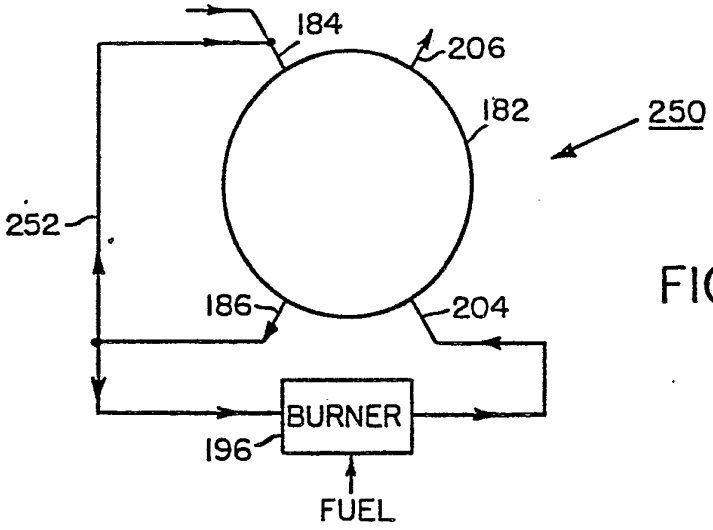


FIG. 20

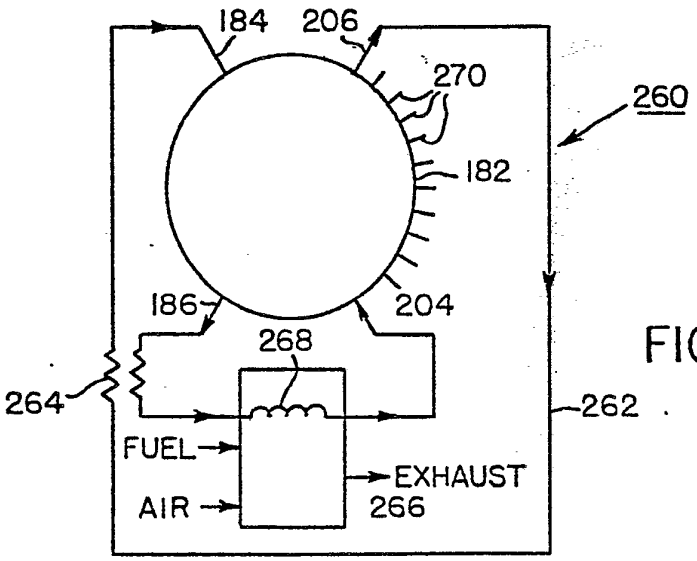


FIG. 21

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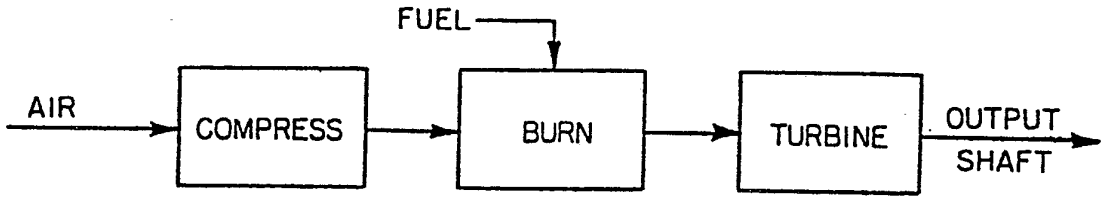


FIG. 22

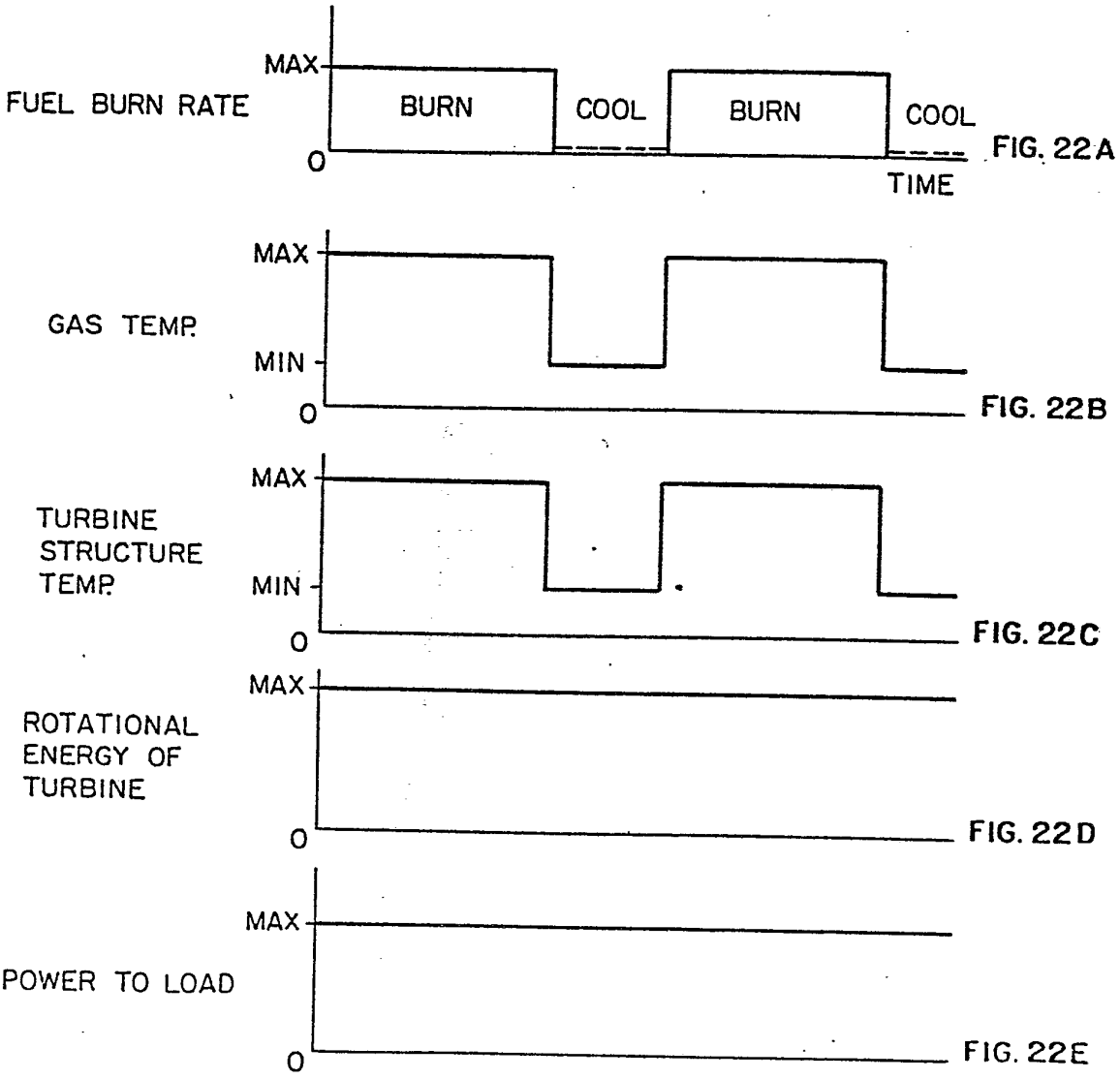


FIG. 23

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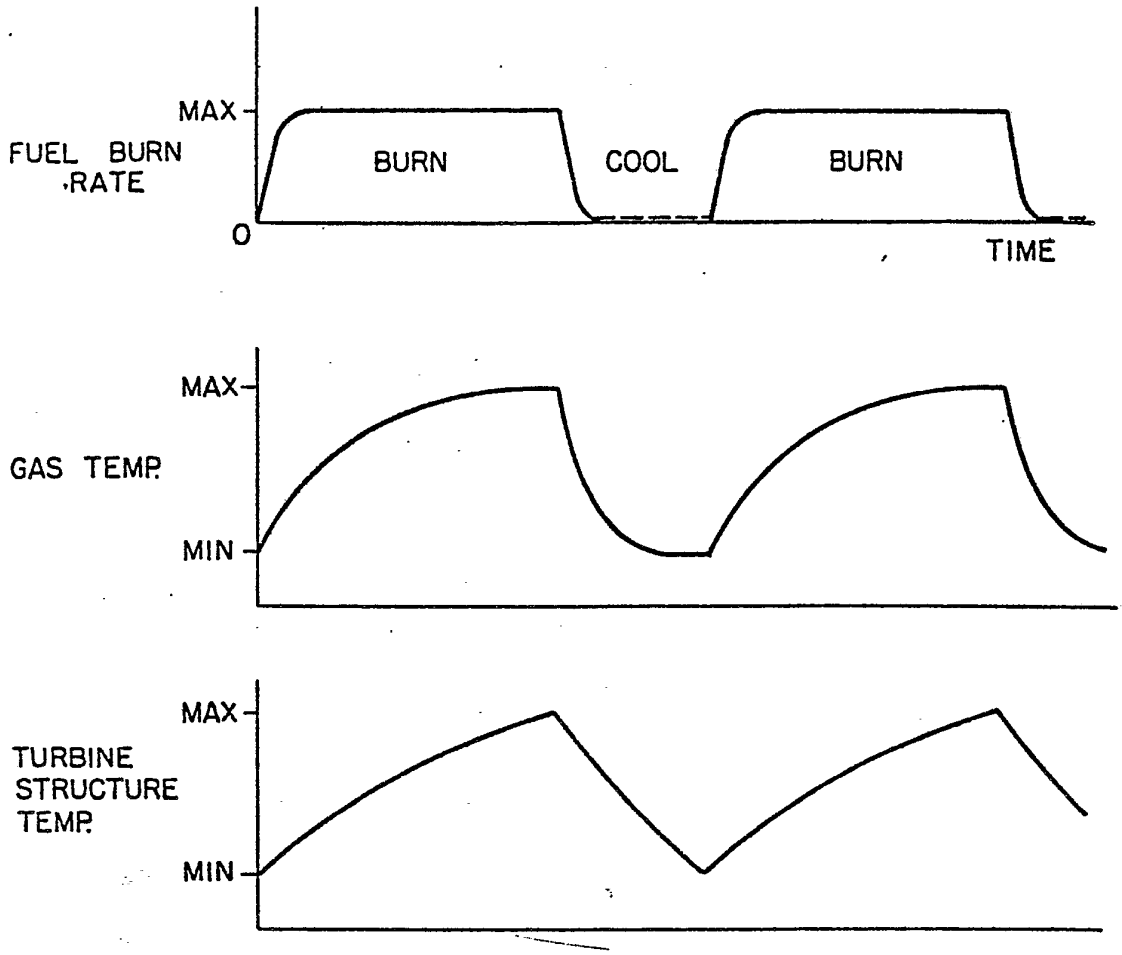


FIG. 24

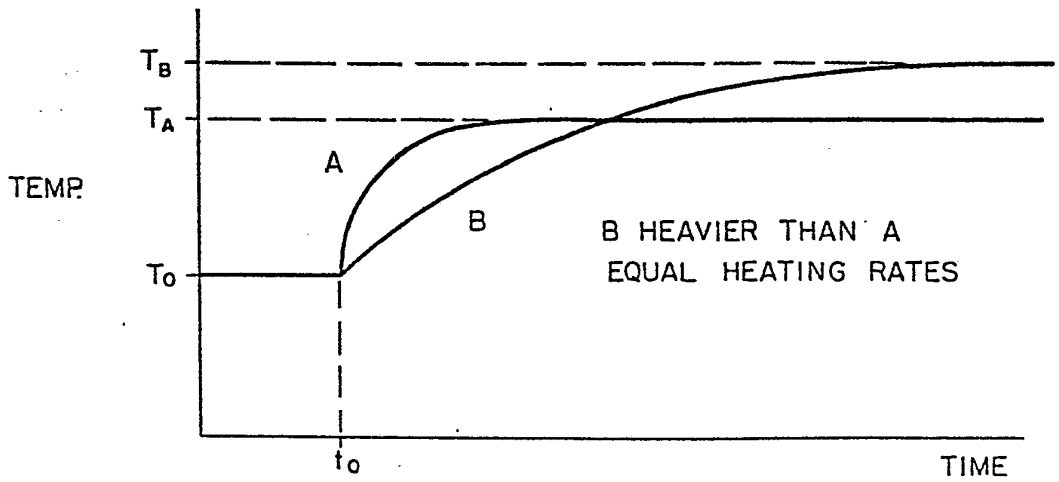


FIG. 25

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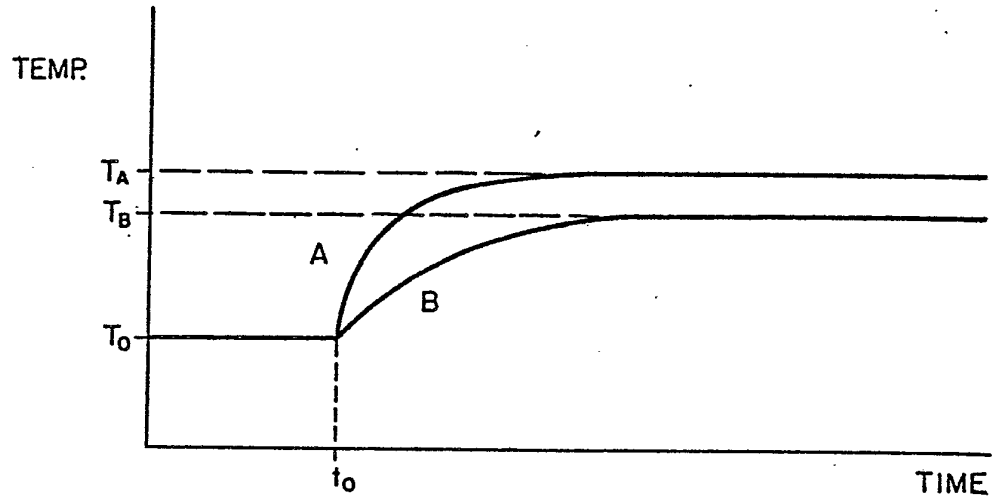


FIG. 26

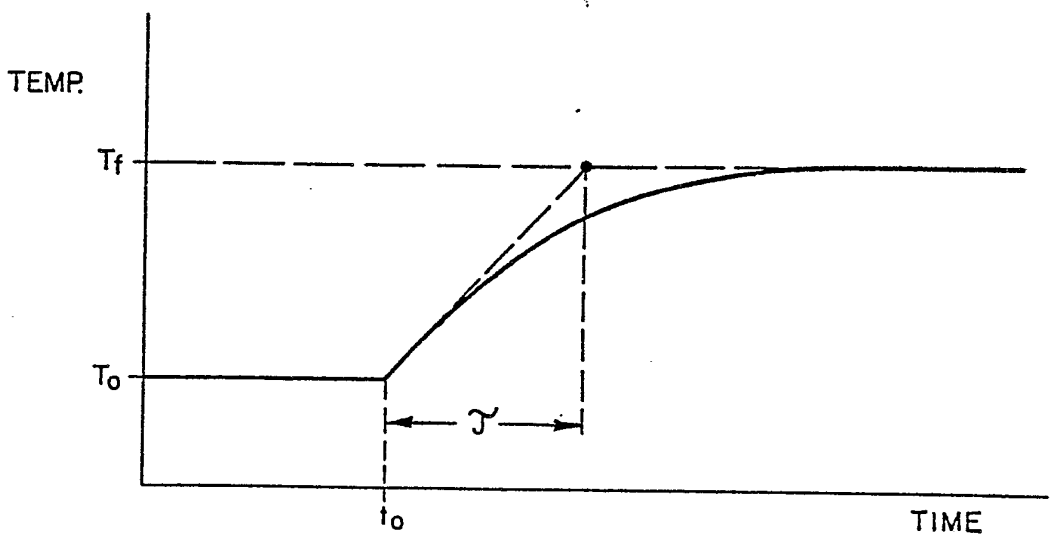


FIG. 27

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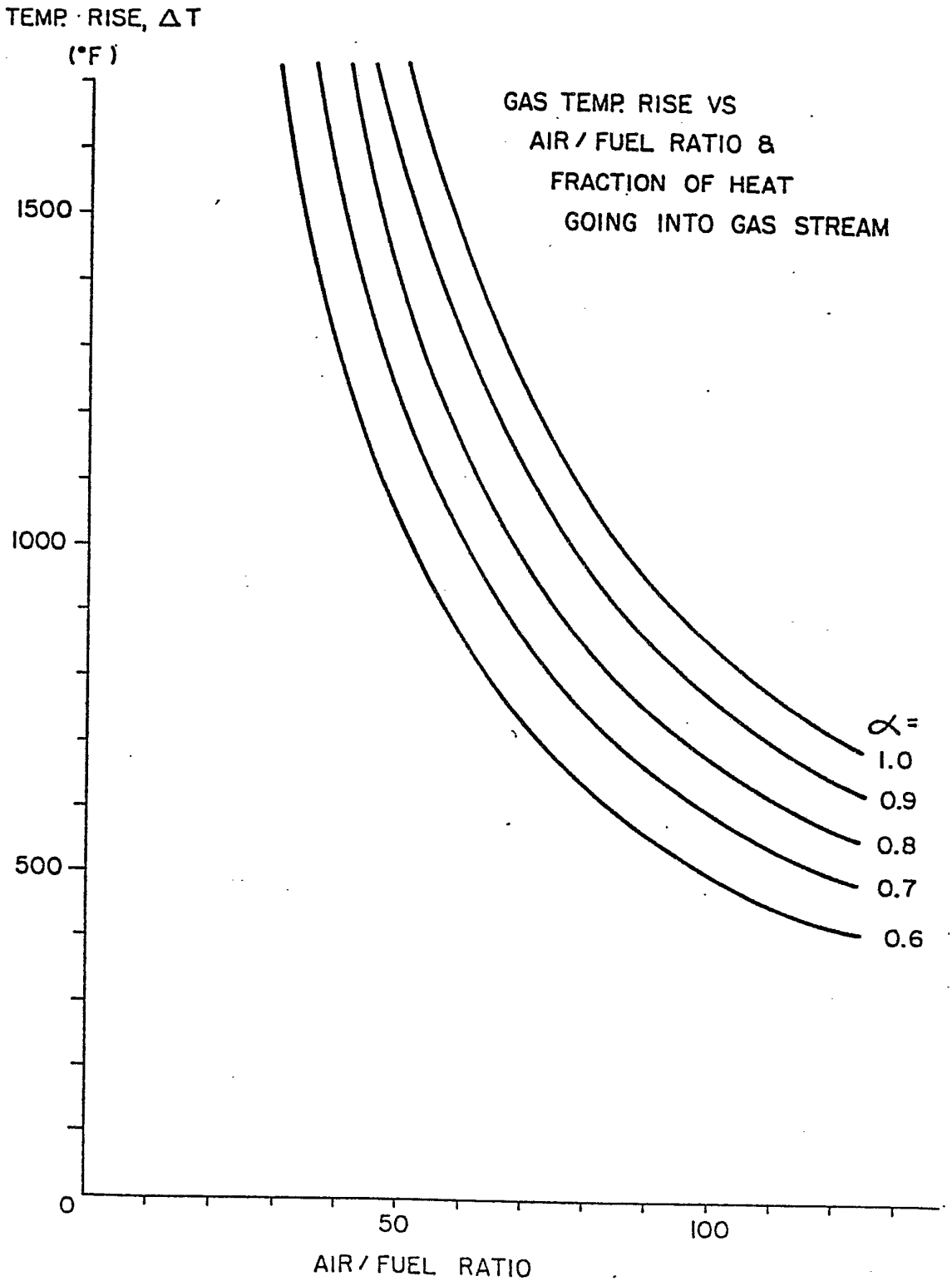


FIG. 28

BUREAU OMPI WIPO

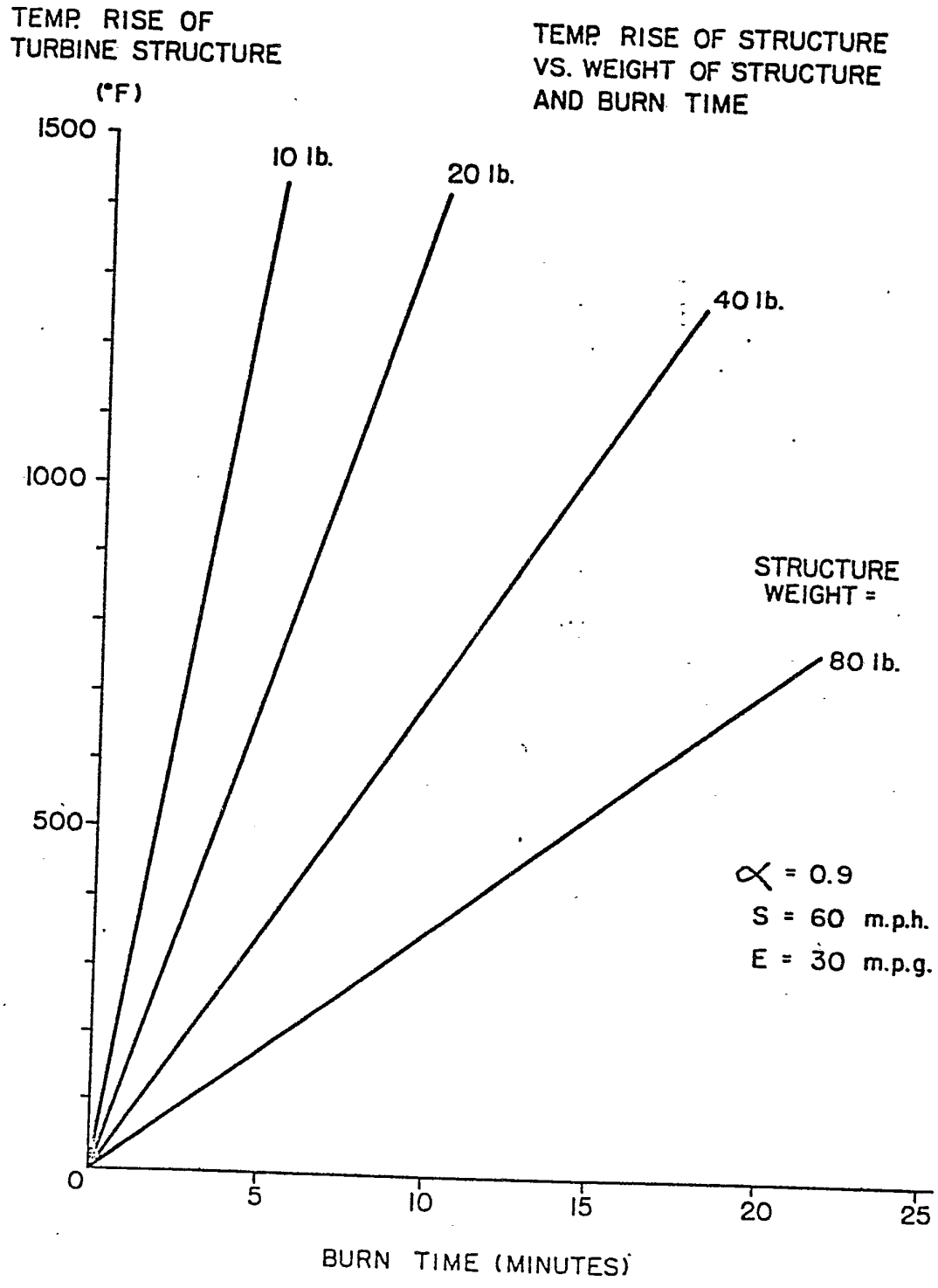


FIG. 29

BUREAU
OMPI
WIPO

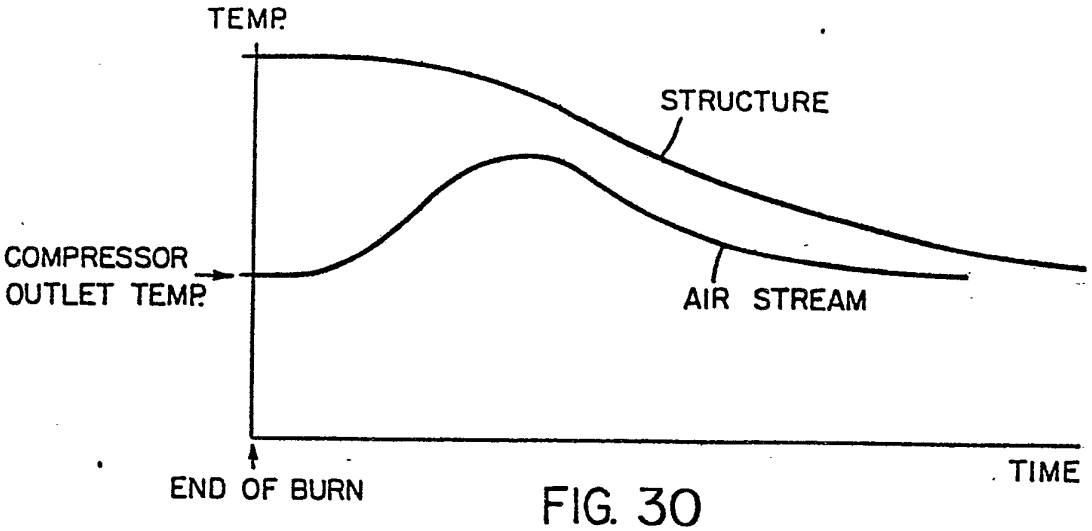


FIG. 30

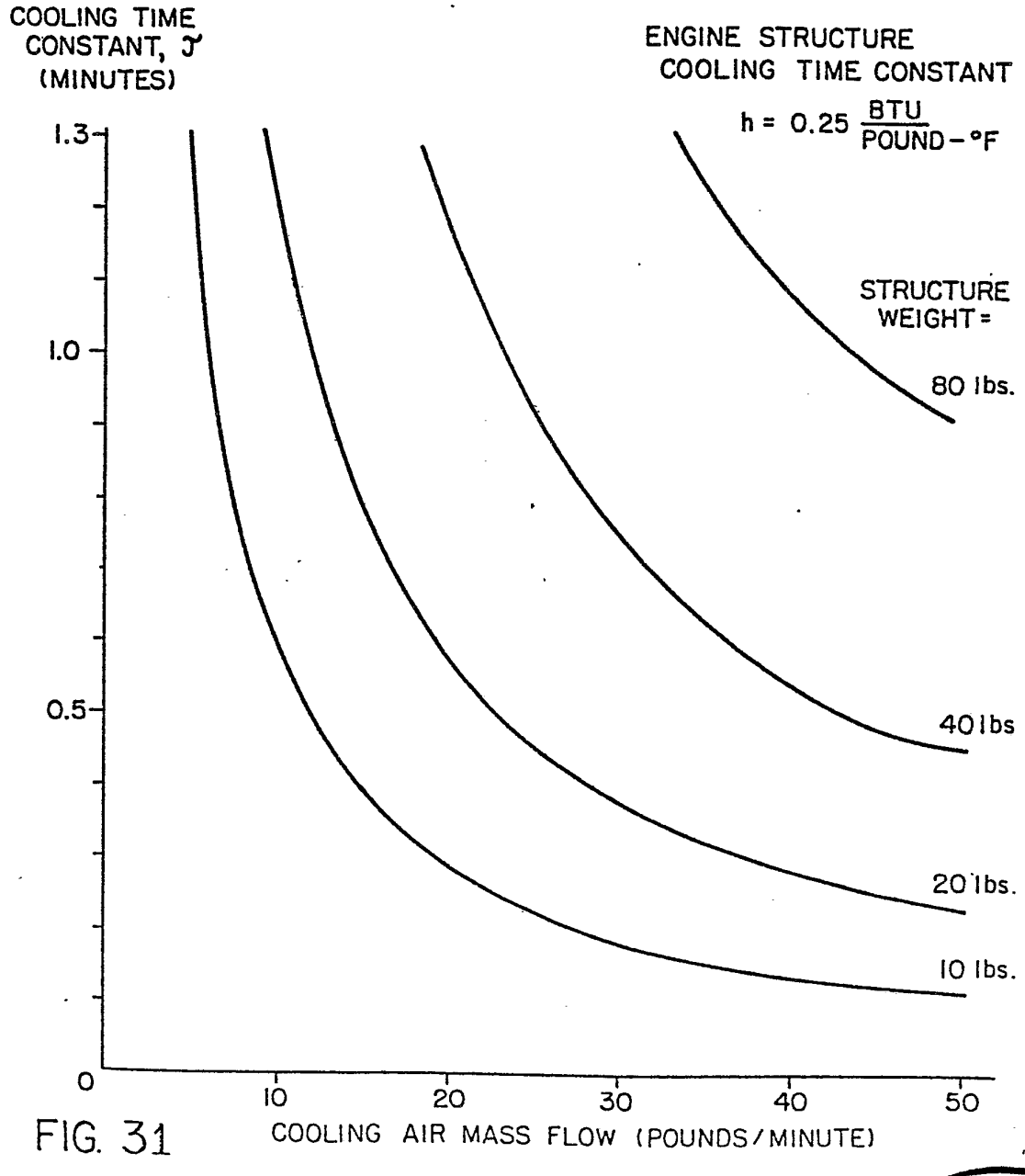


FIG. 31

BUREAU OMPI

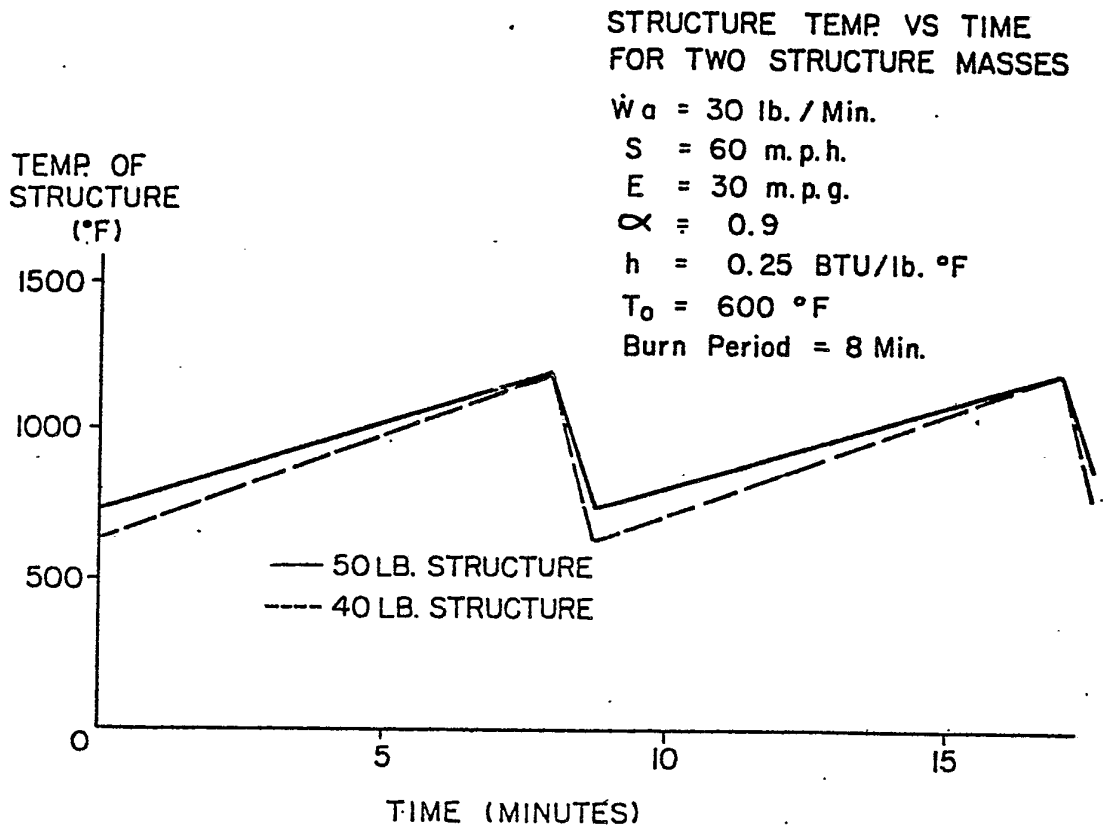


FIG. 32

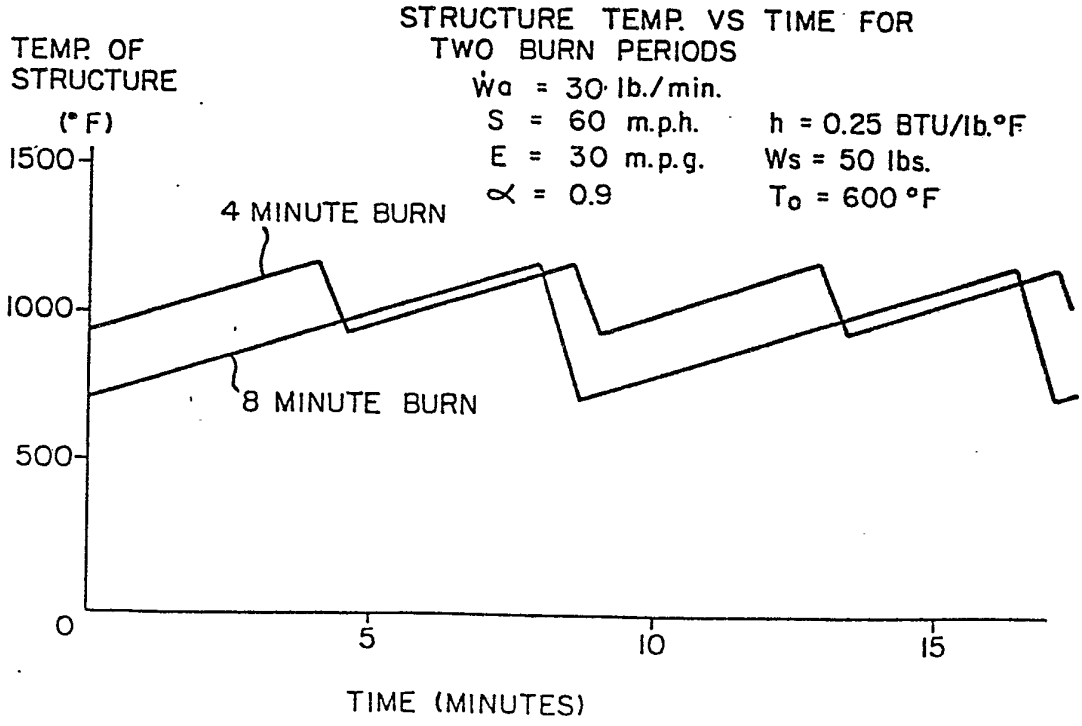


FIG. 33

BUREAU OMPI

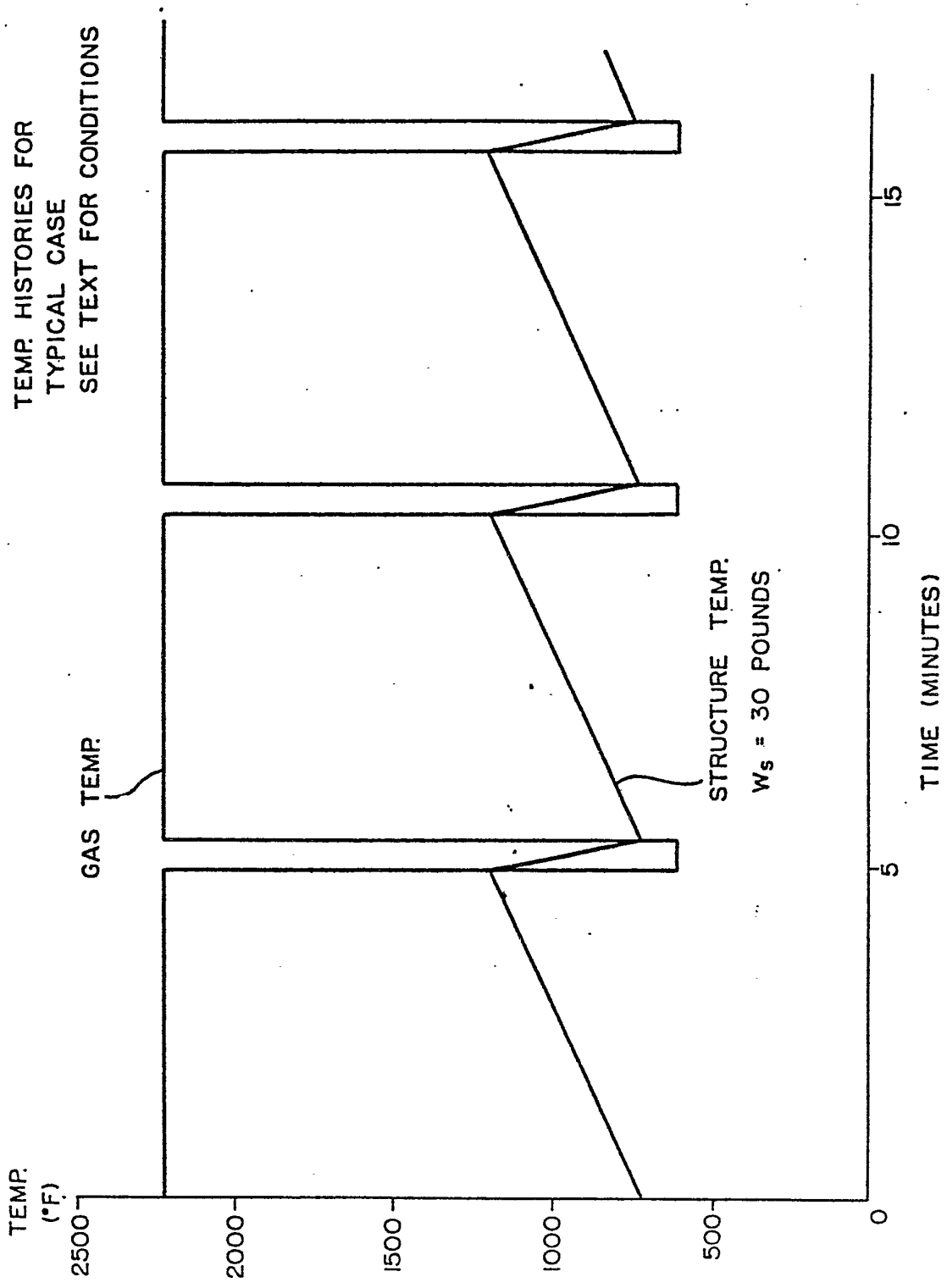


FIG. 34

BUREAU
OMPI

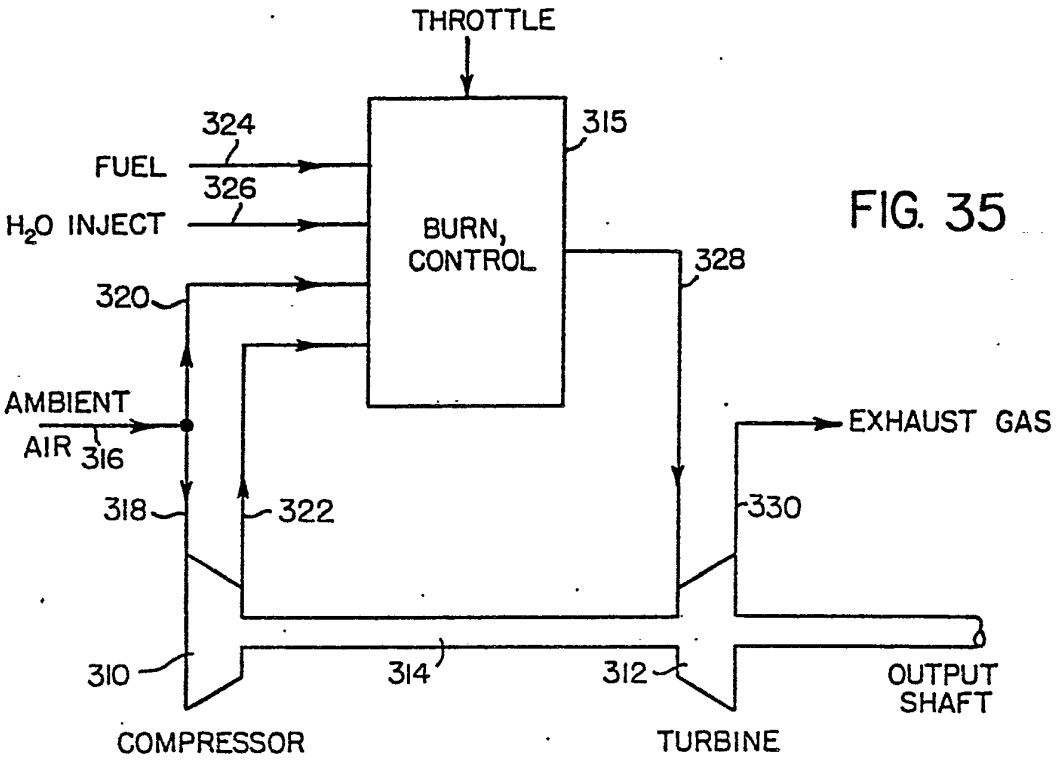


FIG. 35

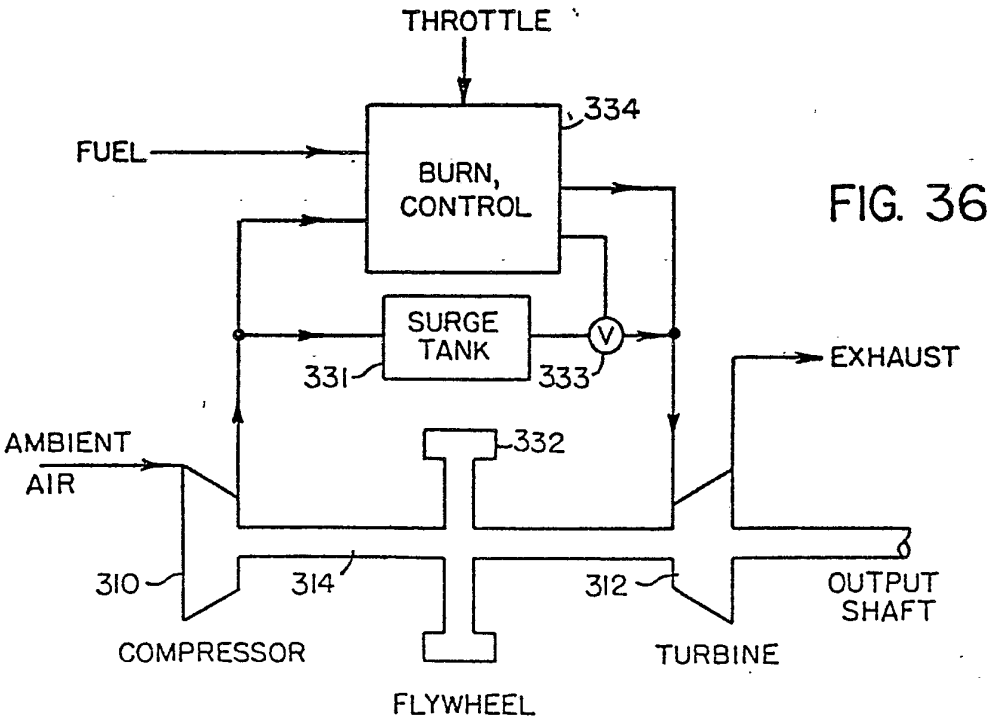


FIG. 36

BUREAU OMPI

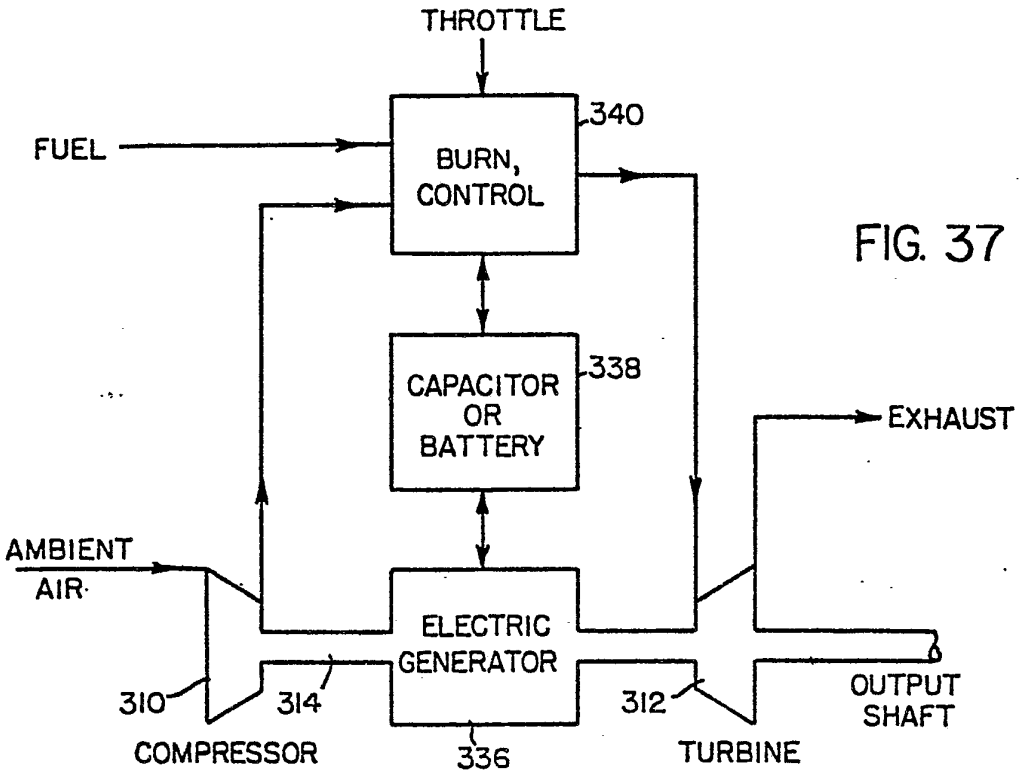


FIG. 37

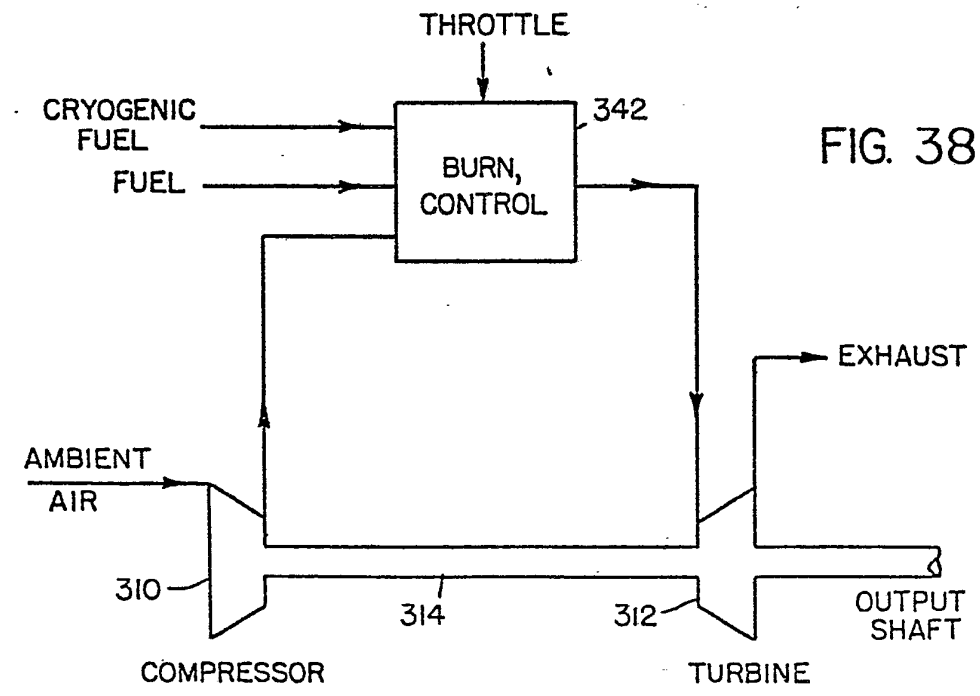


FIG. 38

BUREAU

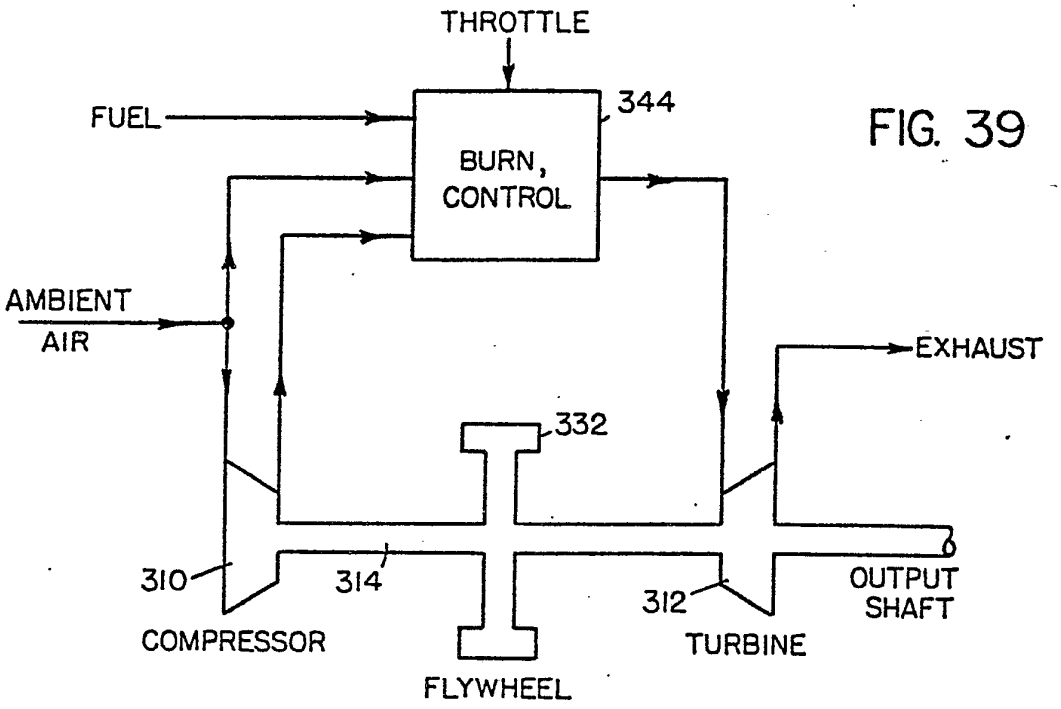


FIG. 39

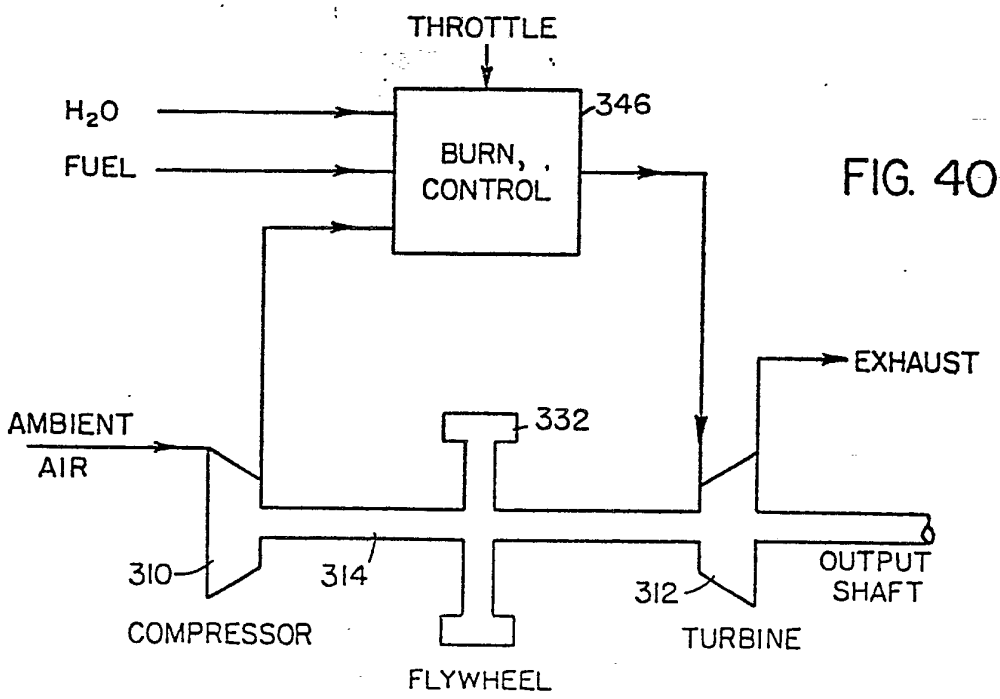


FIG. 40

BUREAU

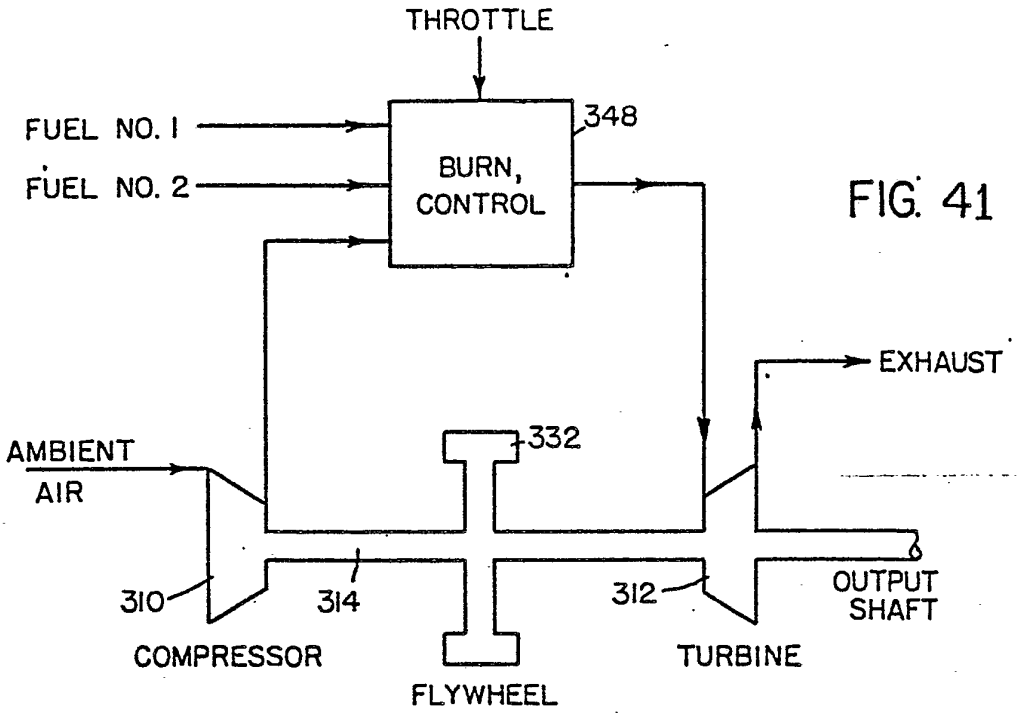


FIG. 41

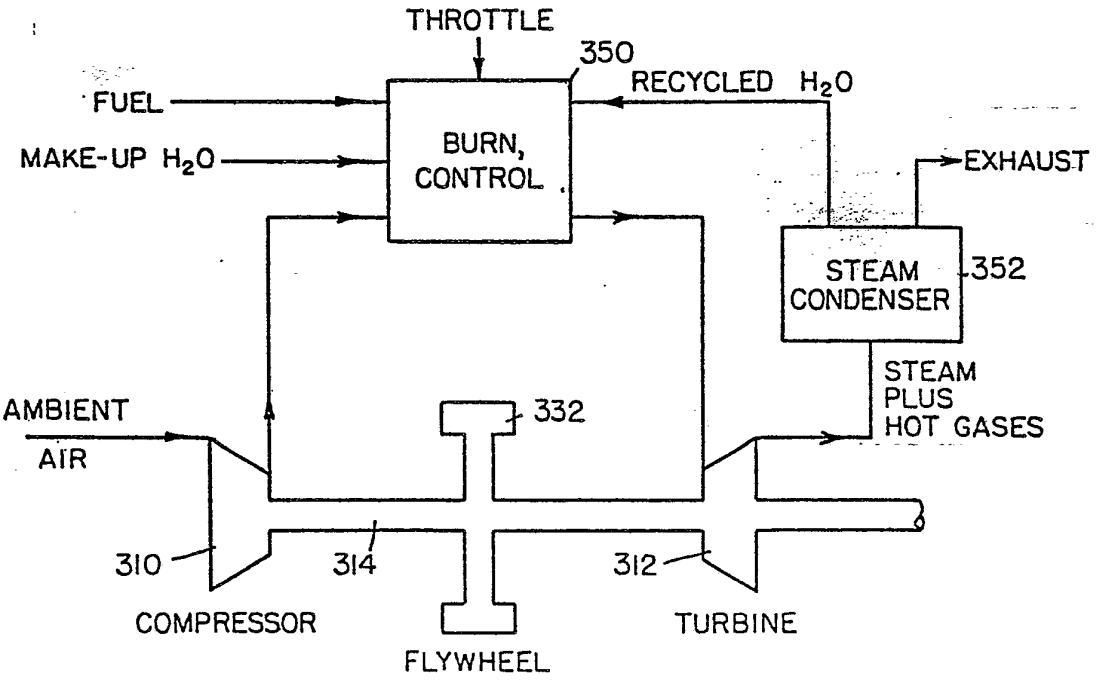


FIG. 42

BUREAU

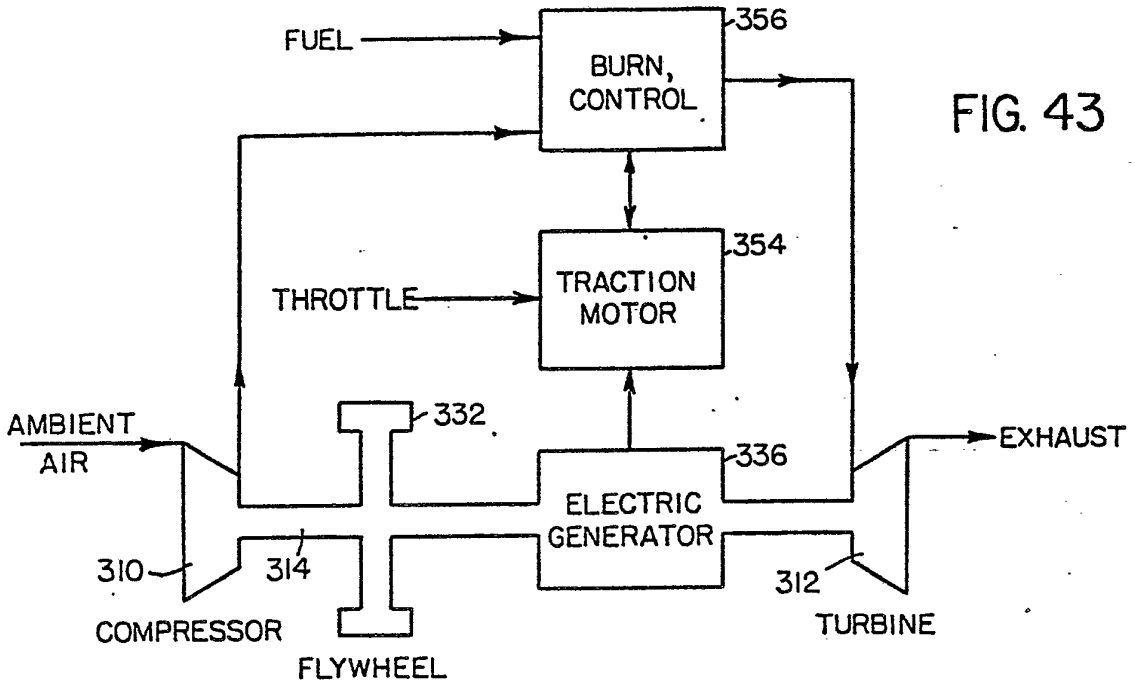


FIG. 43

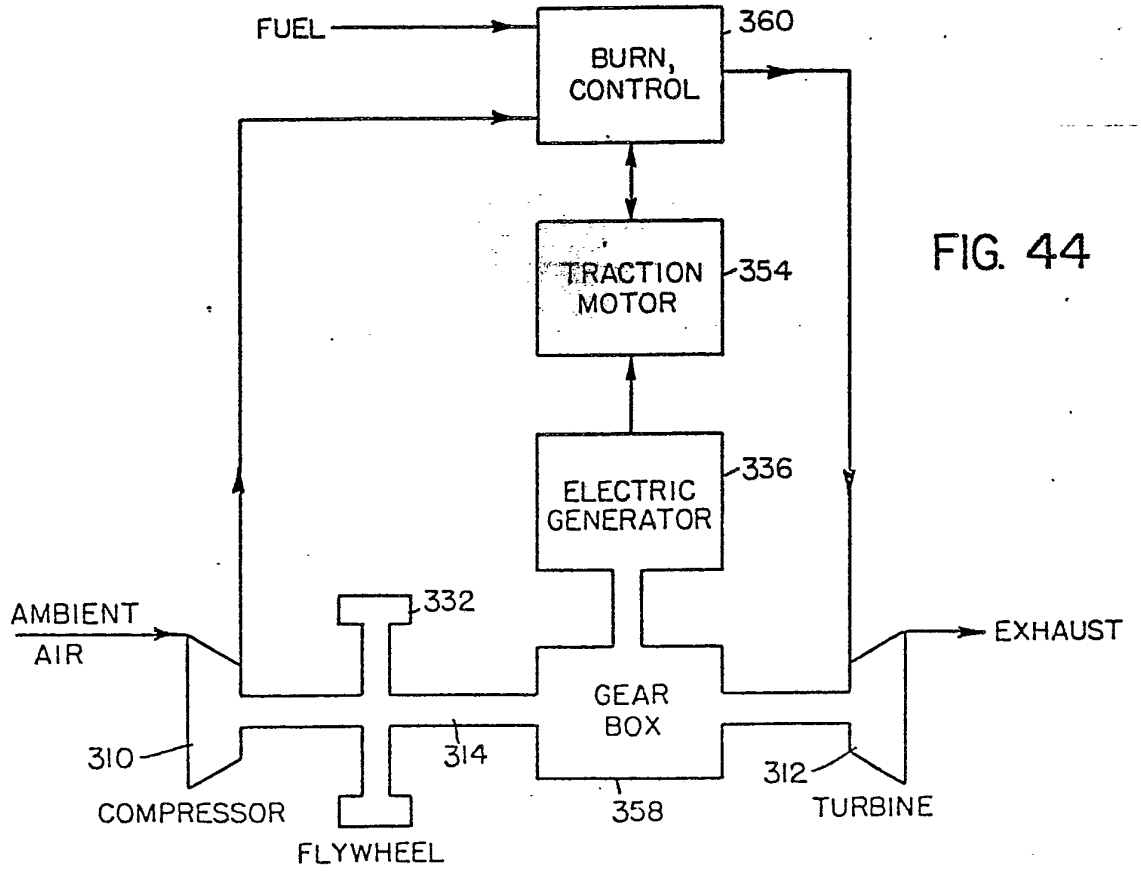


FIG. 44

BUREAU OMPI WIPO

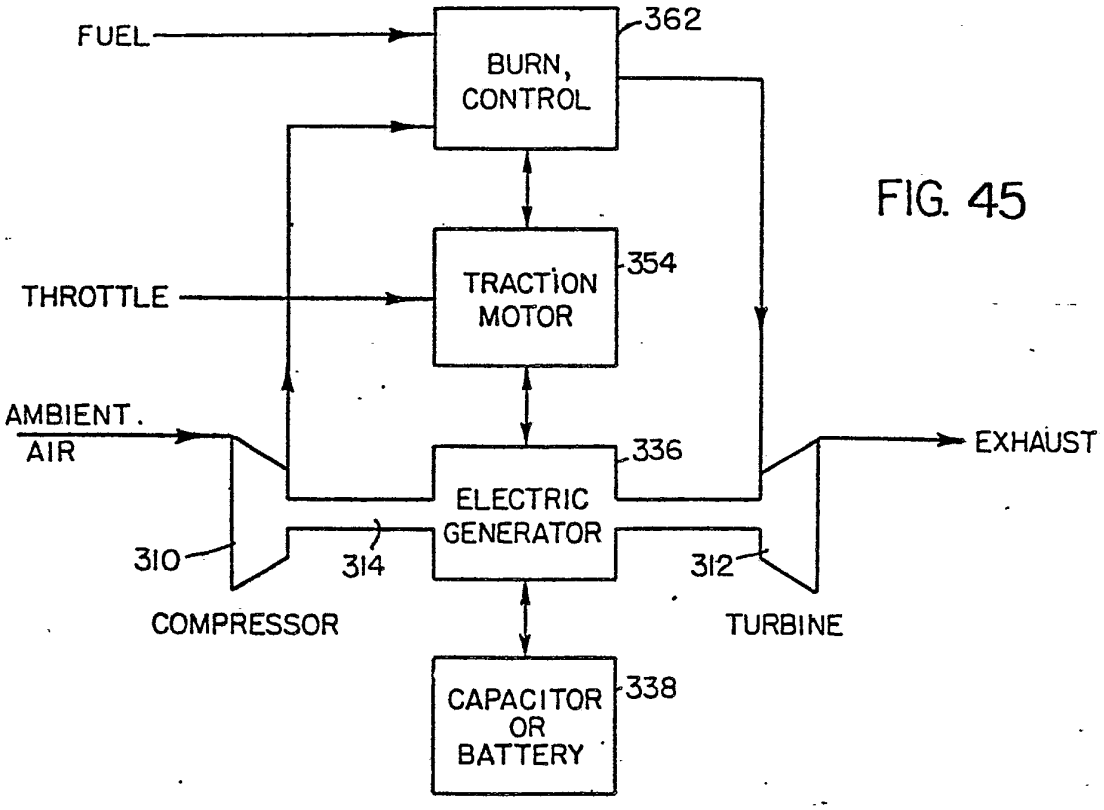


FIG. 45

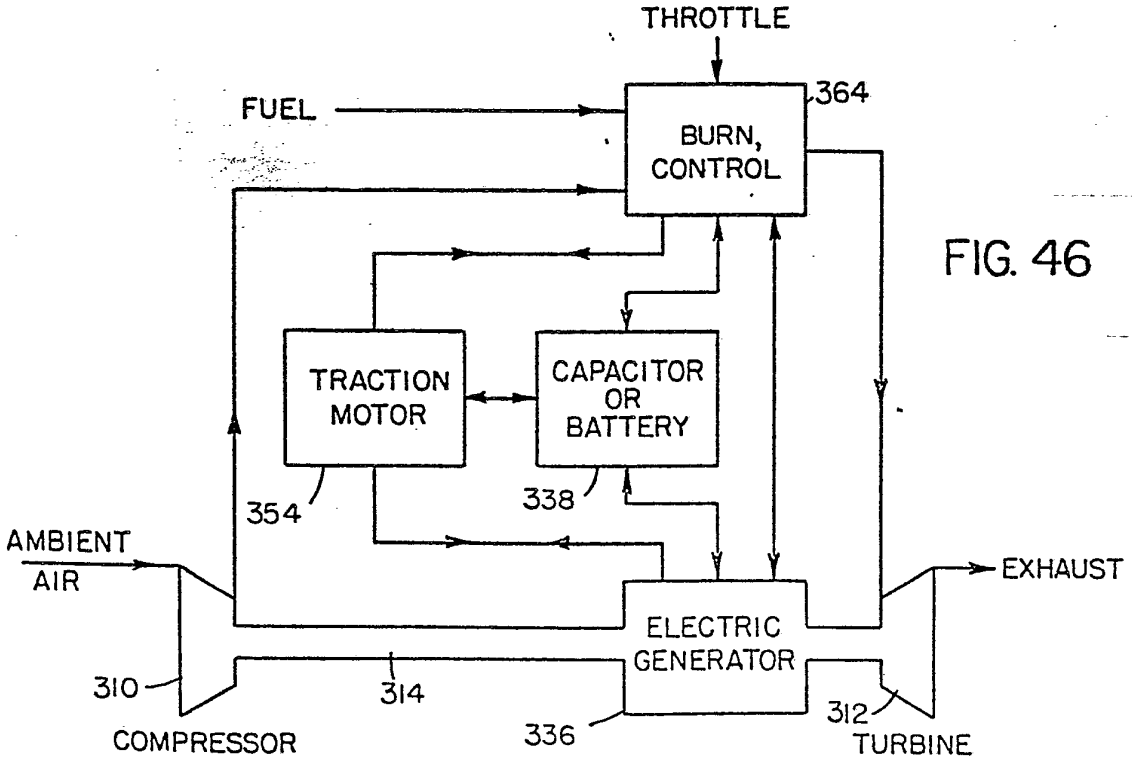


FIG. 46

BUREAU OMPI

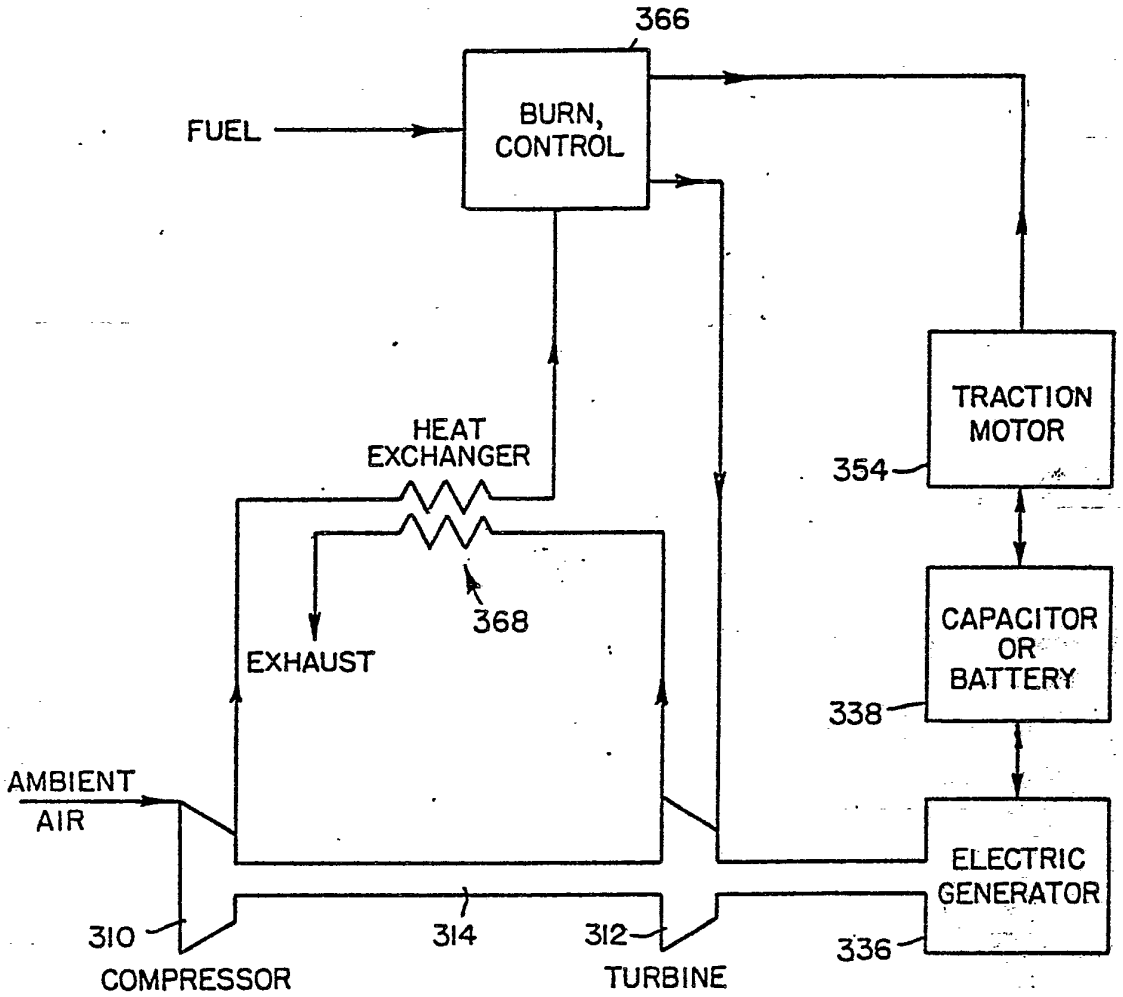
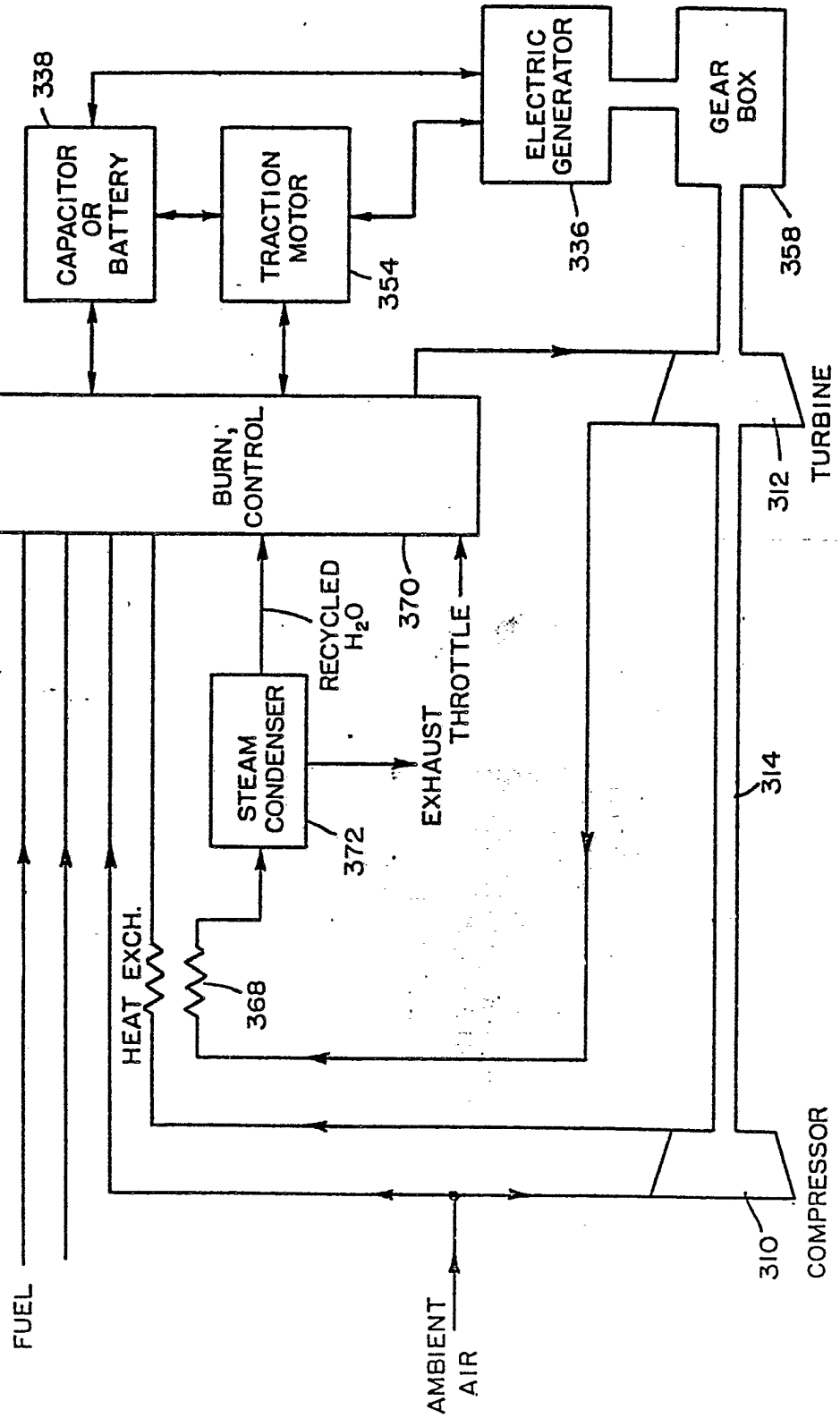


FIG. 47

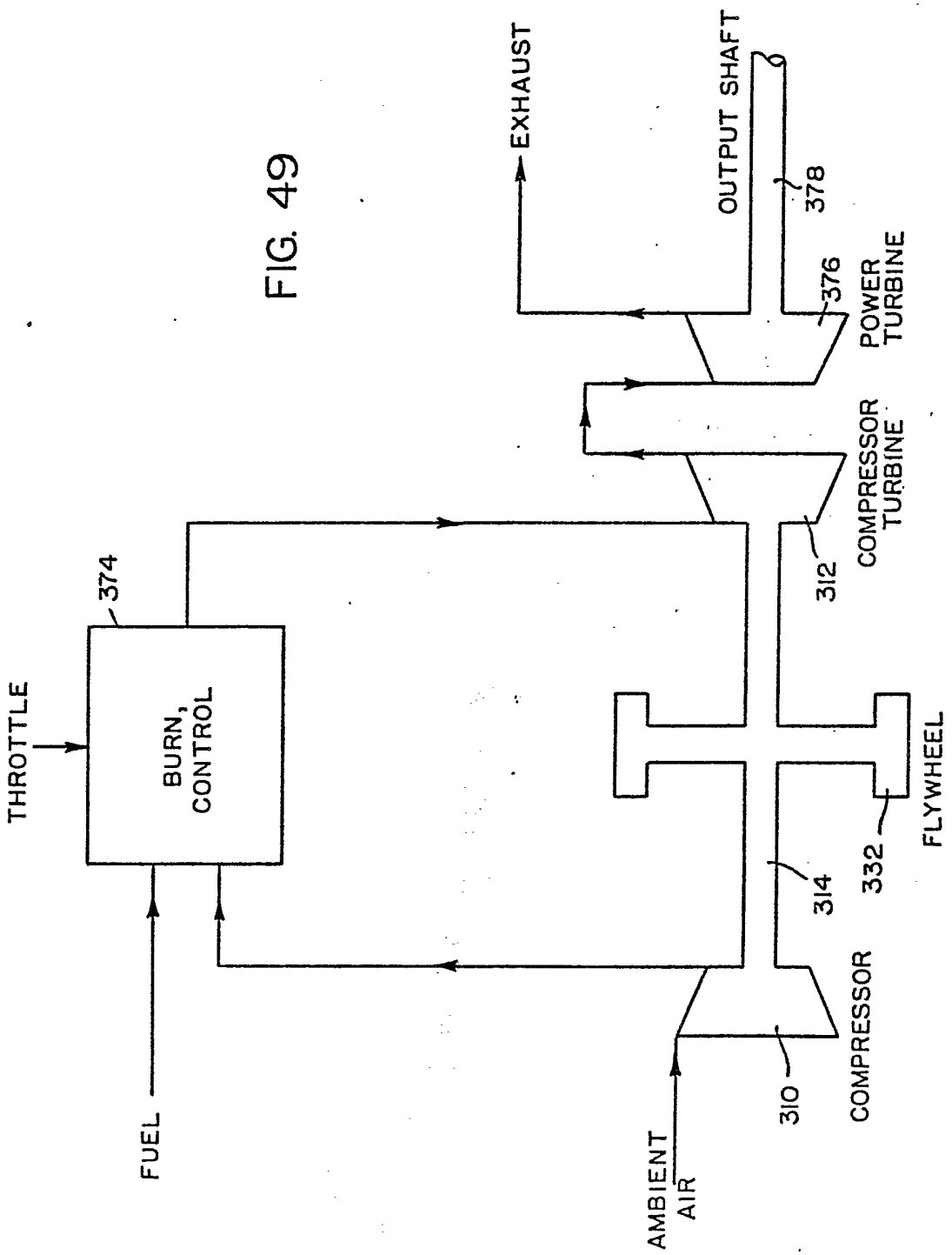
BUREAU OMPI WIPO

FIG. 48



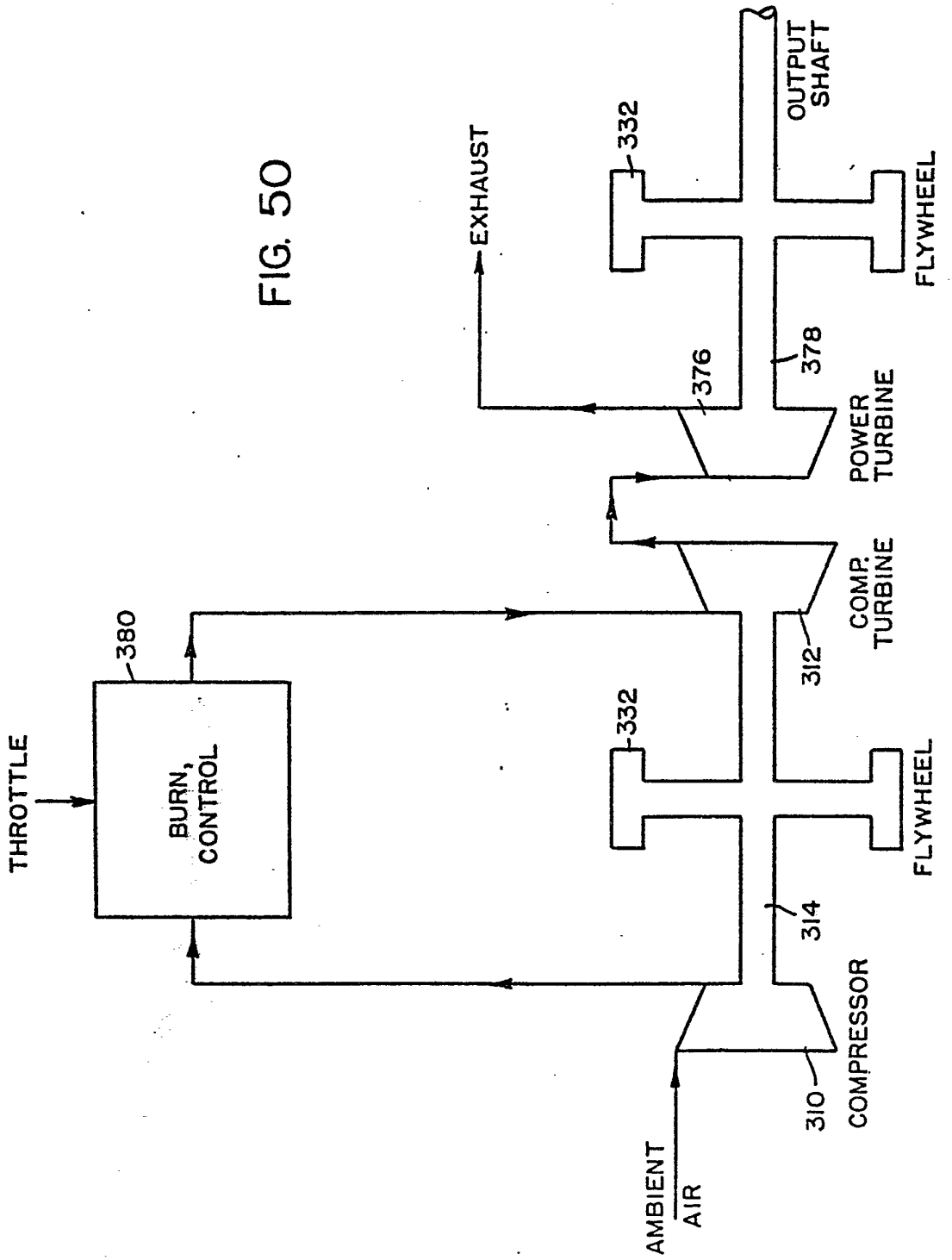
BUREAU OMPI WIPO

FIG. 49



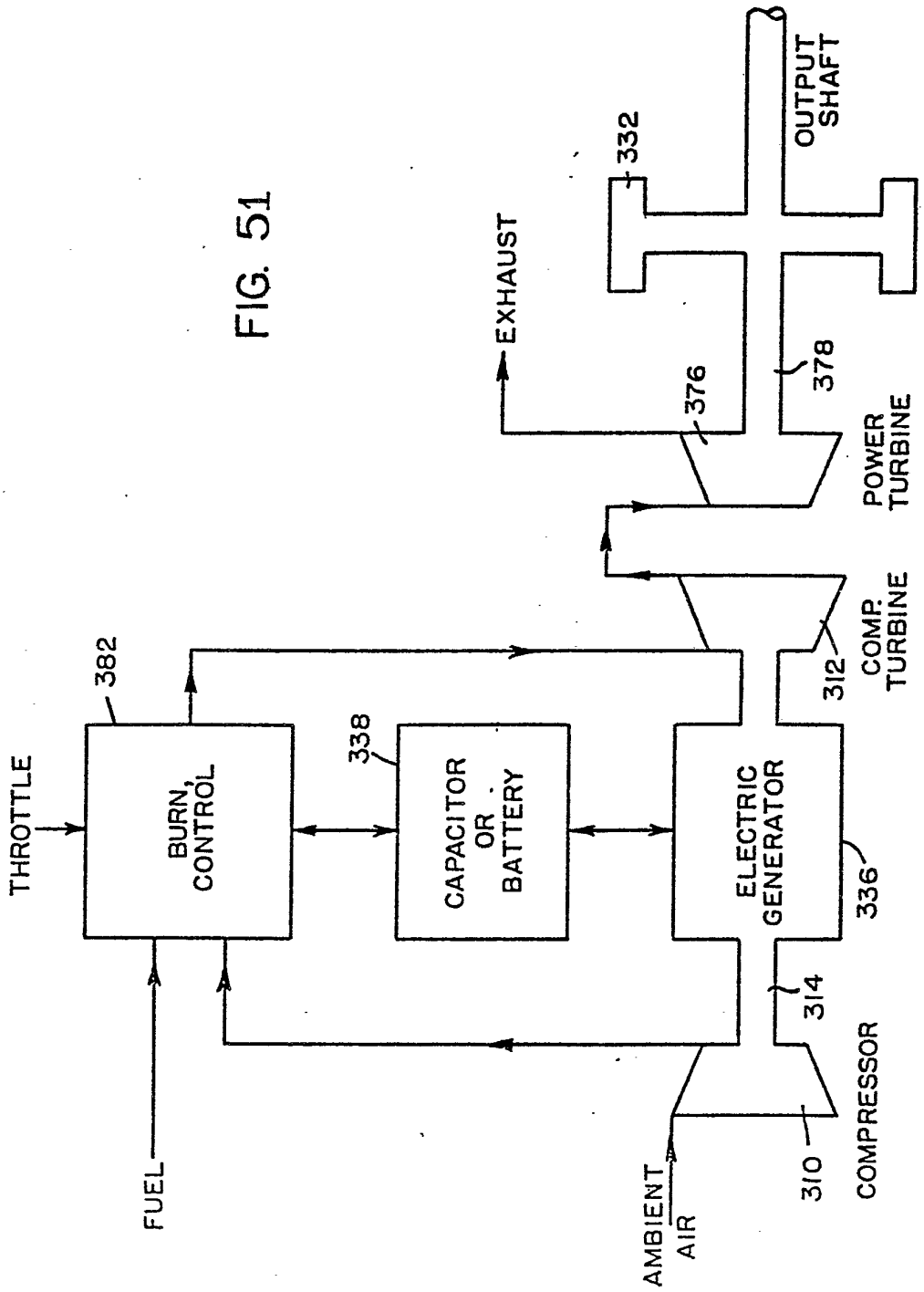
BUREAU

FIG. 50



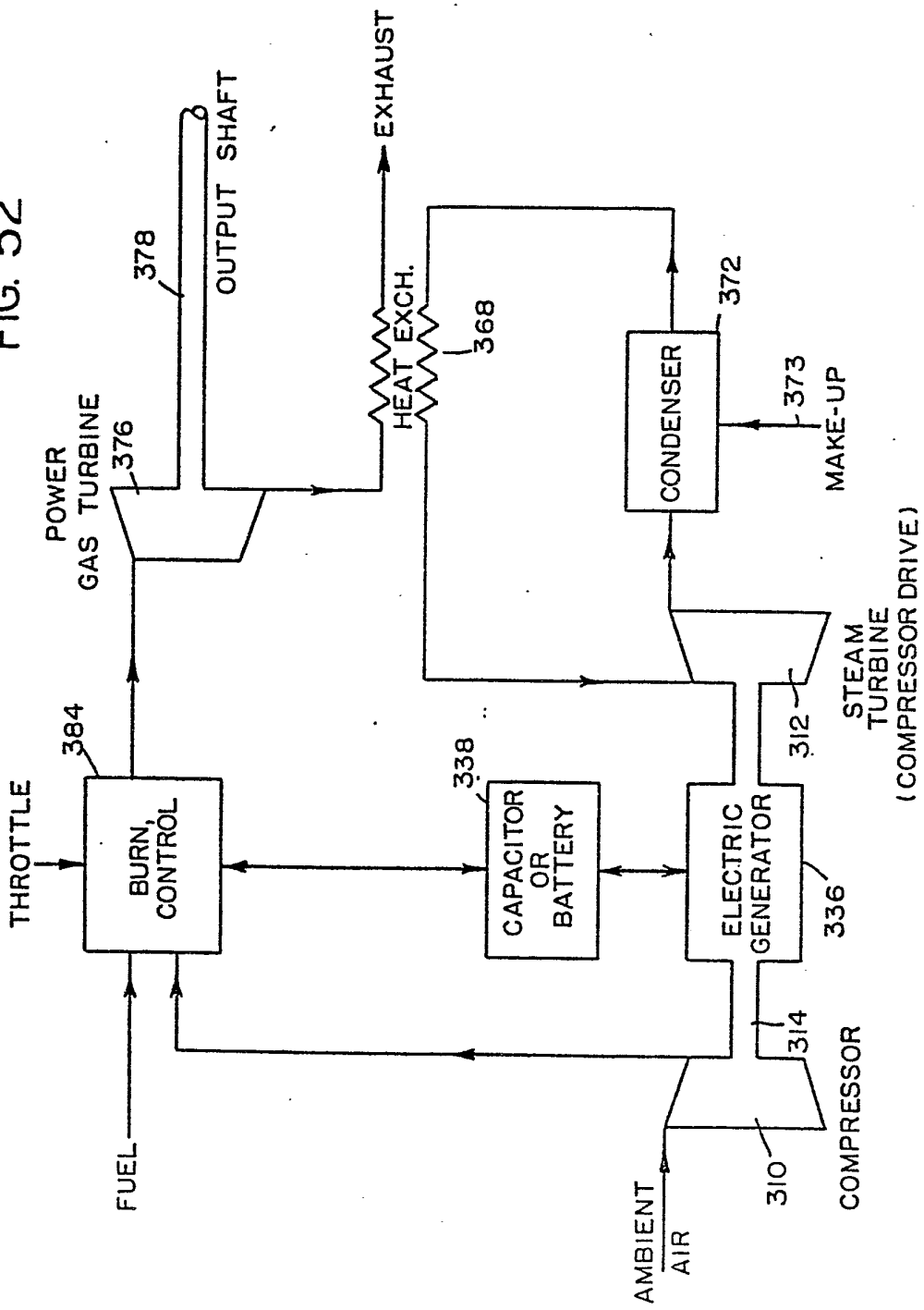
BUREAU

FIG. 51



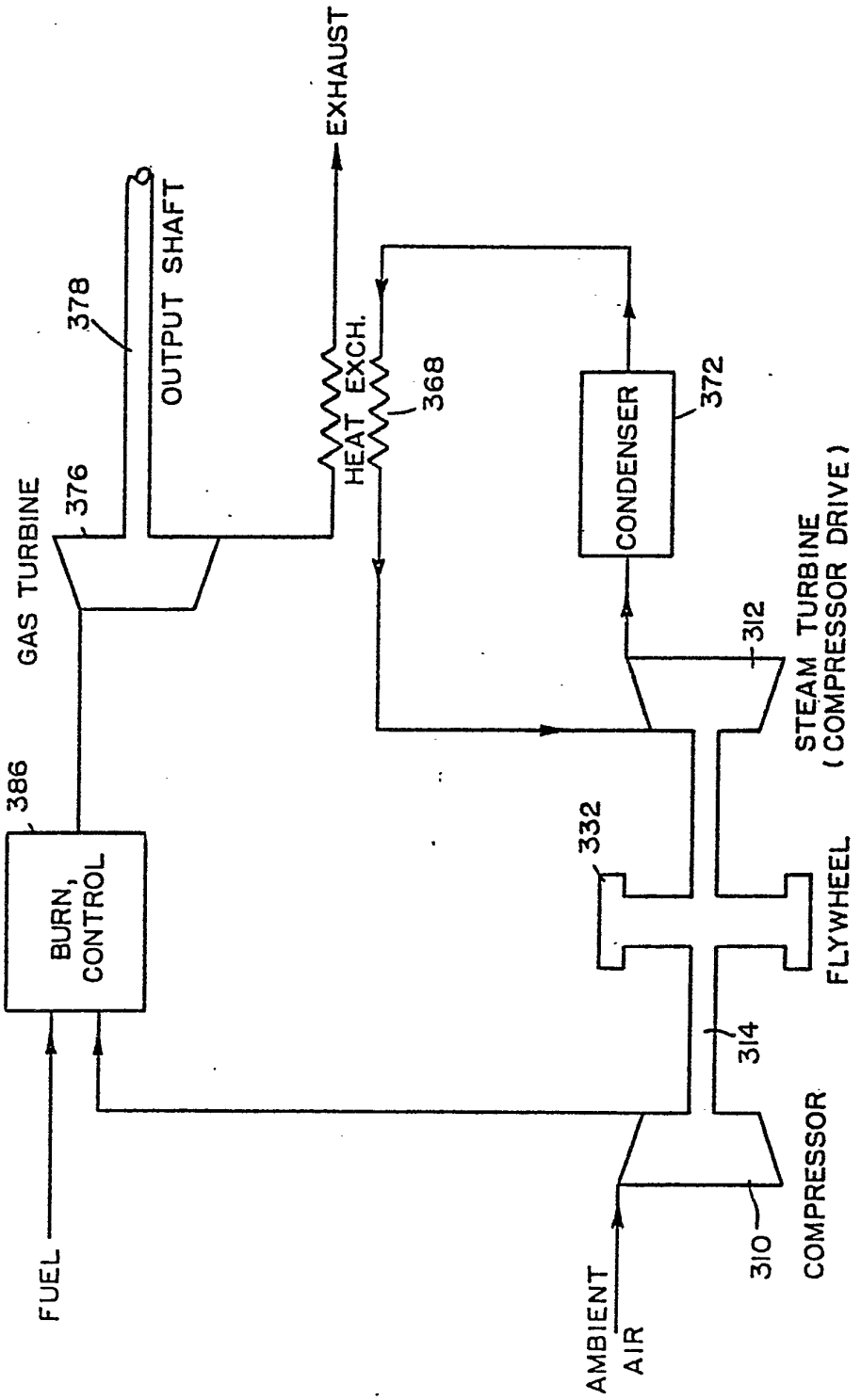
BUREAU OMPI WIPO

FIG. 52



BUREAU OMPI WIPO

FIG. 53



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WIPO

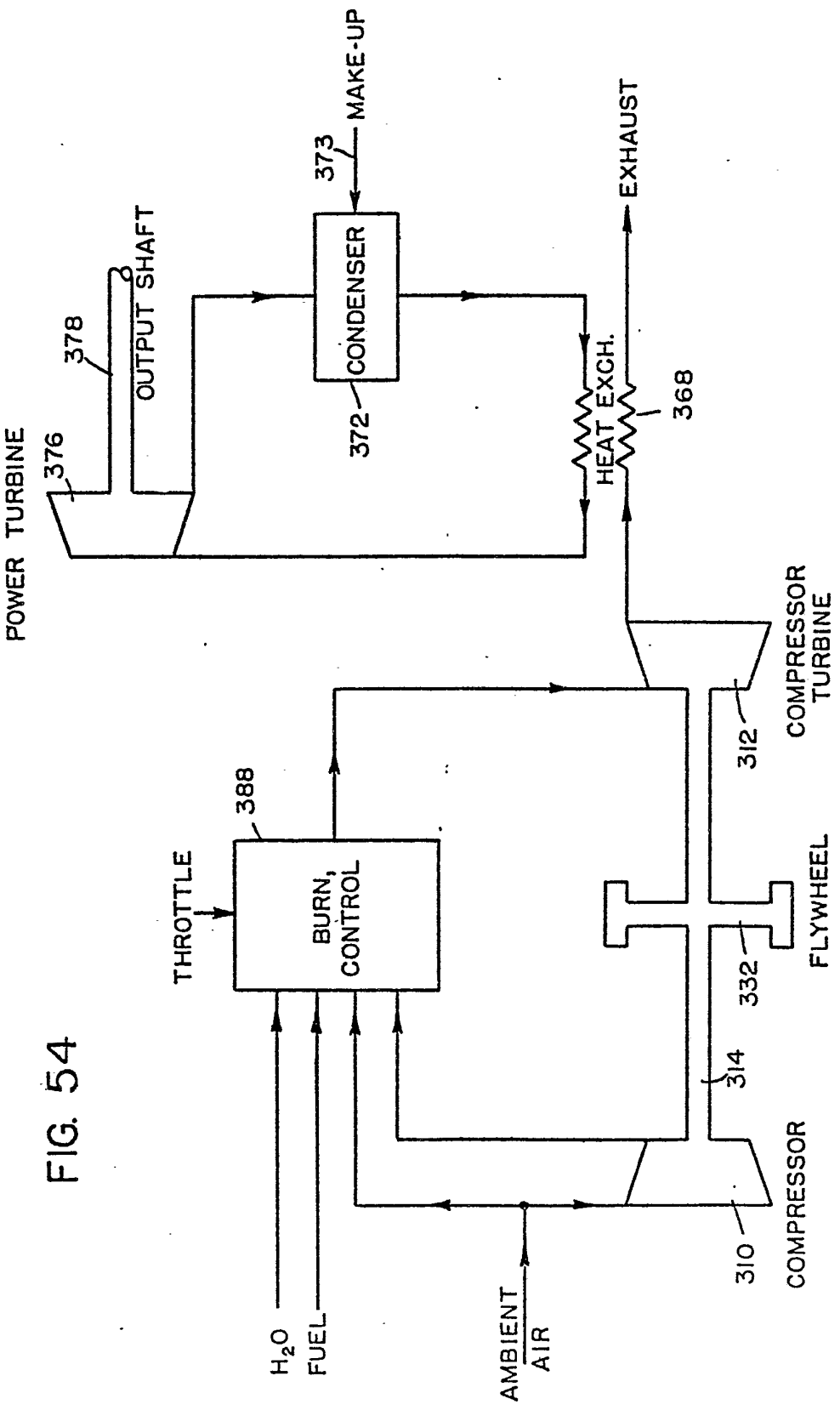
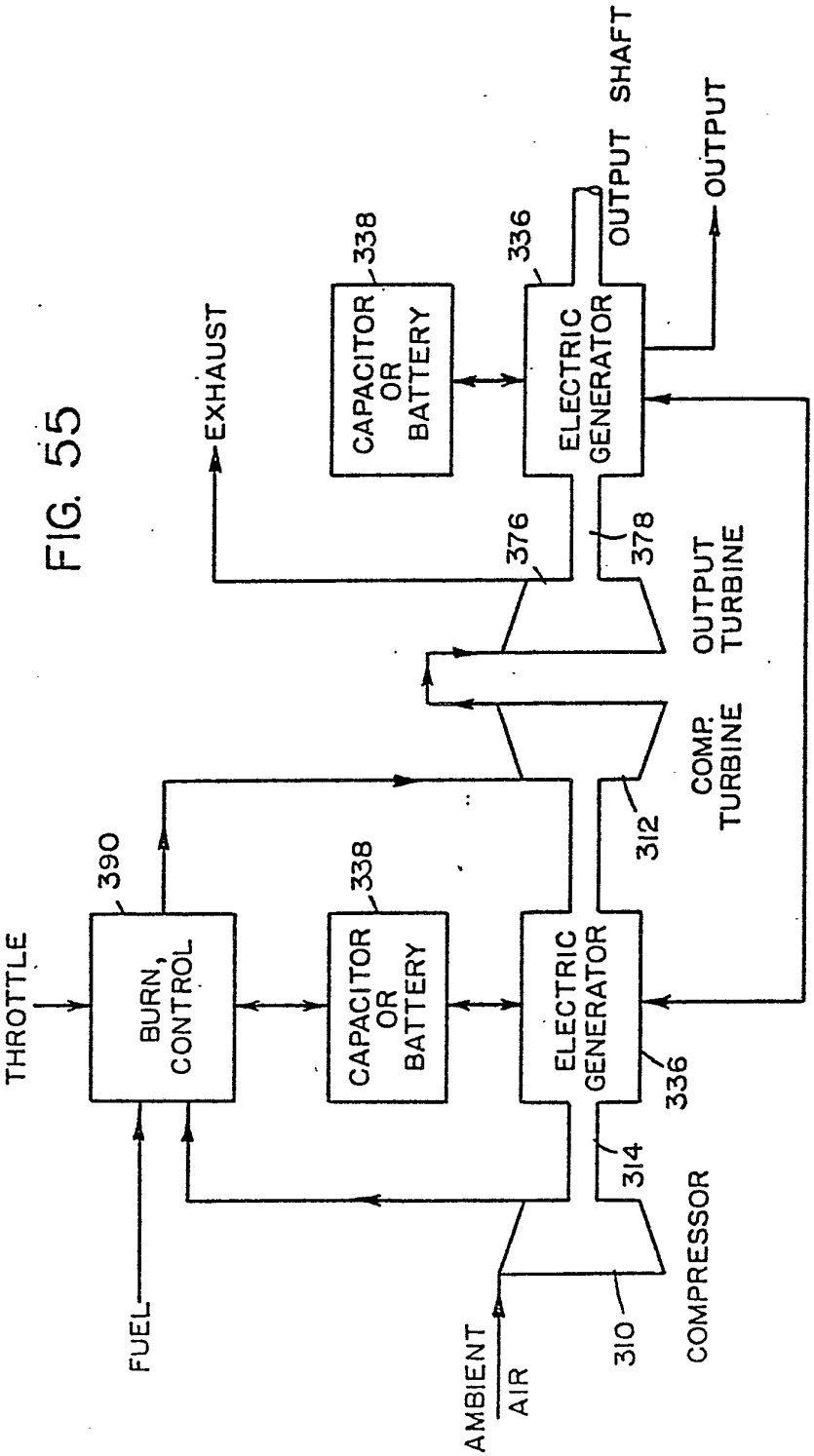


FIG. 54

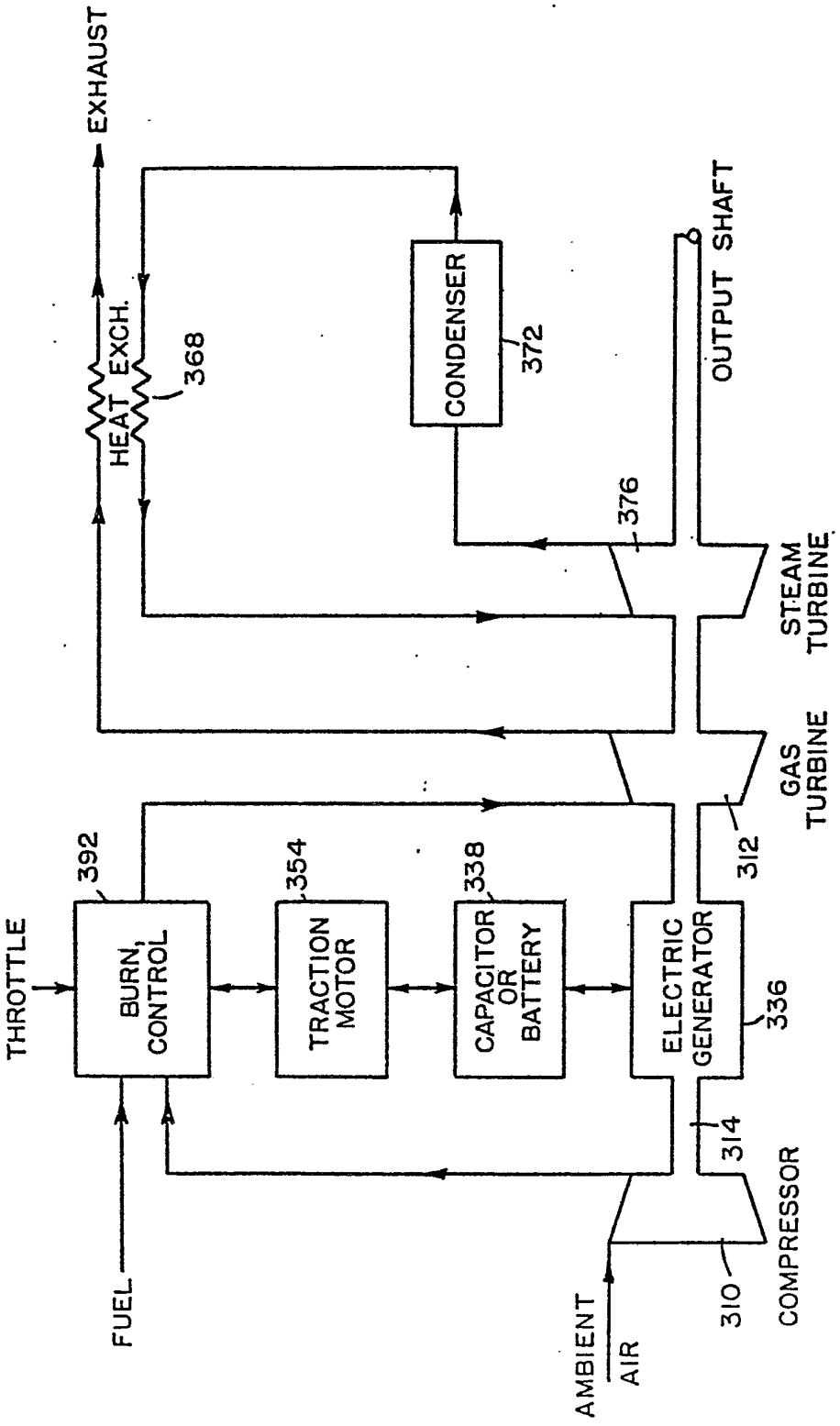
BUREAU

FIG. 55



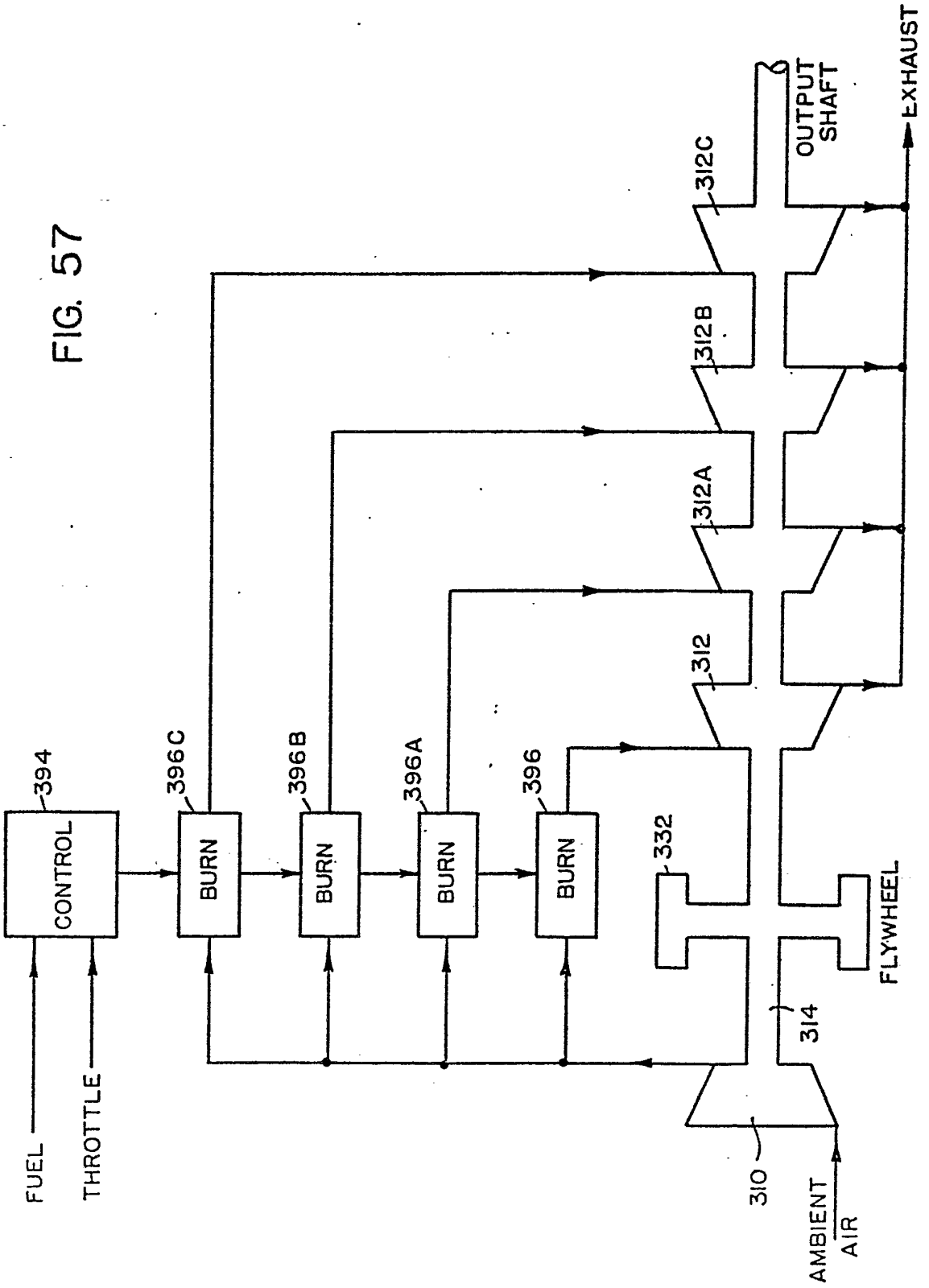
BUREAU

FIG. 56



BUREAU OMPI WIPO

FIG. 57



BUREAU OMPI

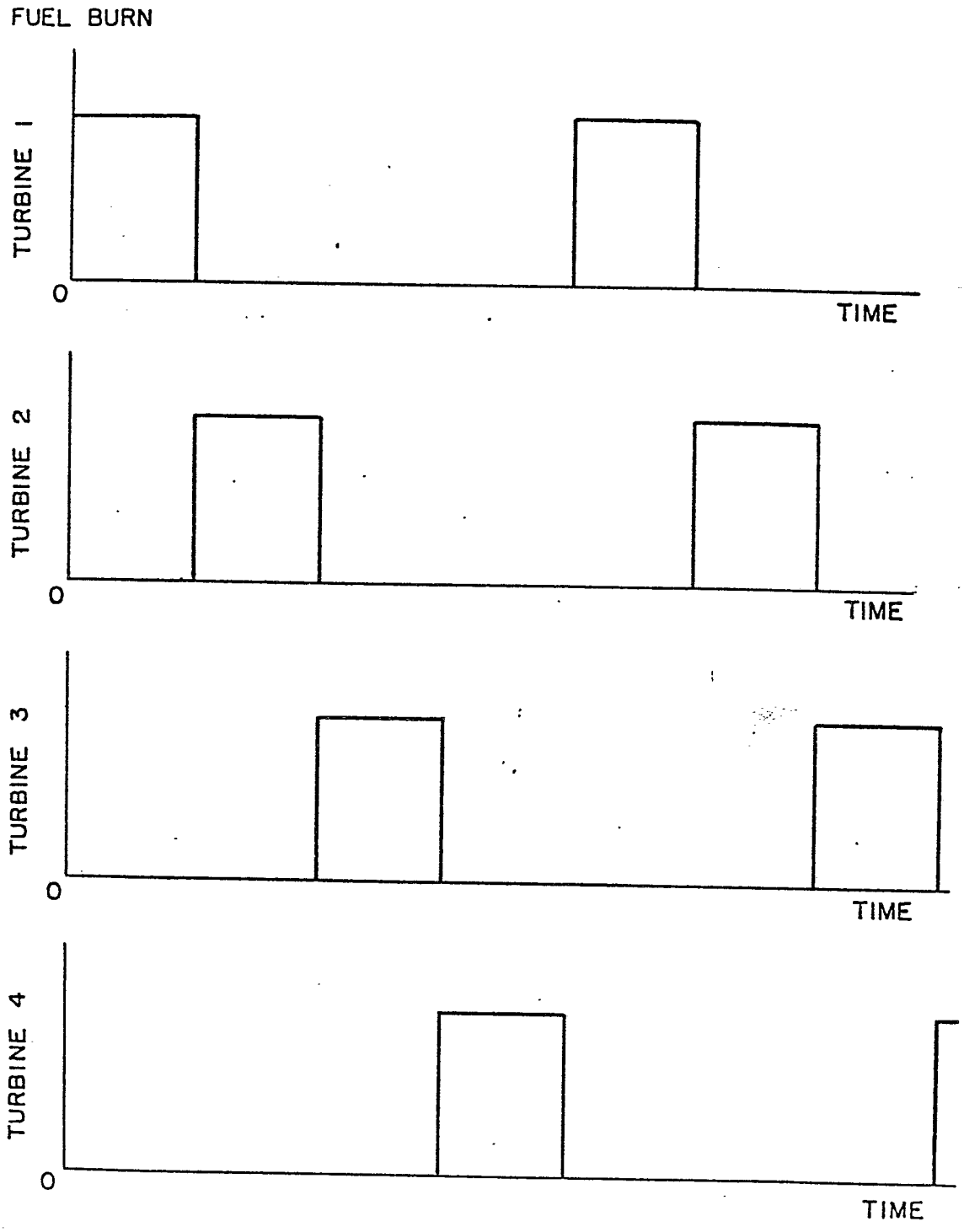


FIG. 58

BUREAU

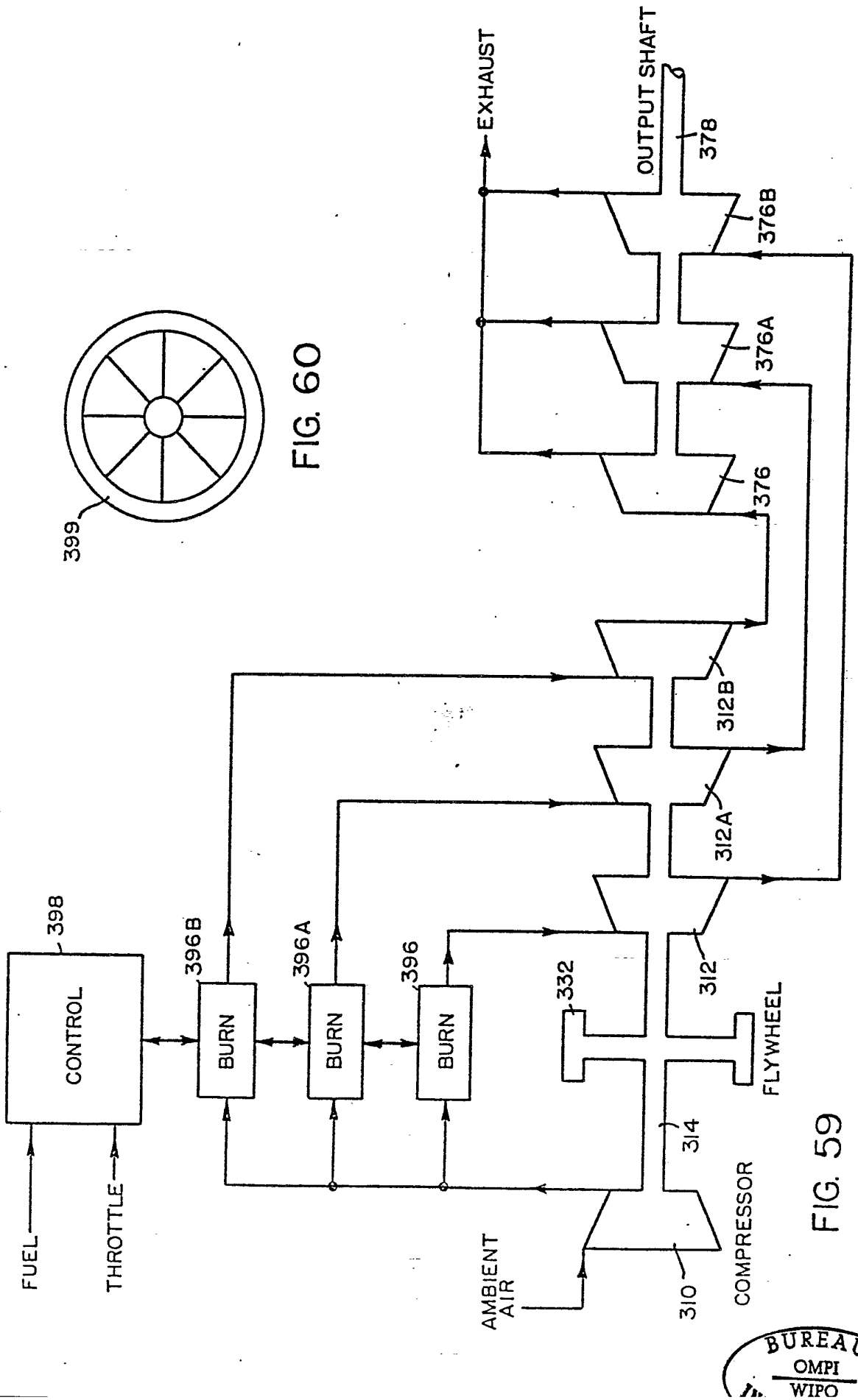


FIG. 59

FIG. 60

BUREAU OMPI WIPO

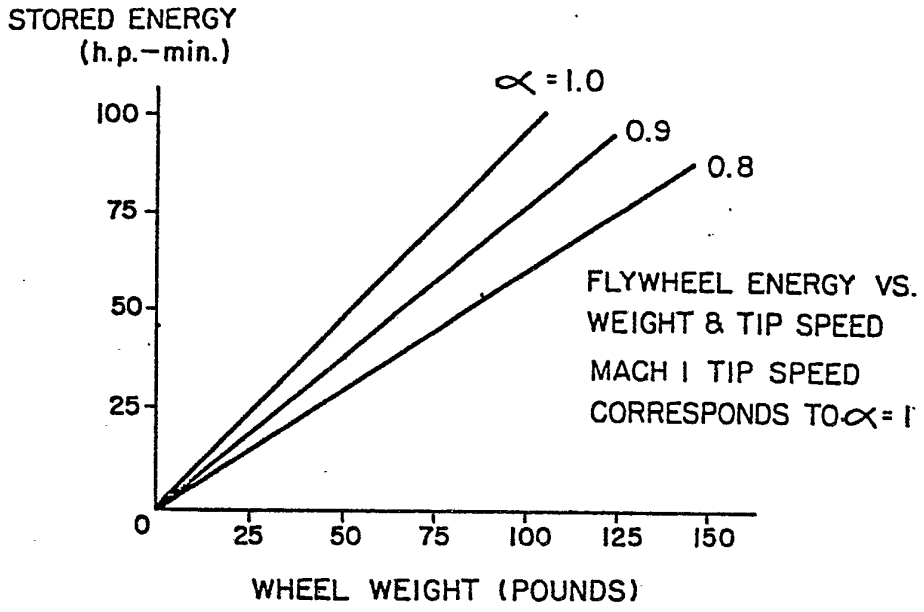


FIG. 61

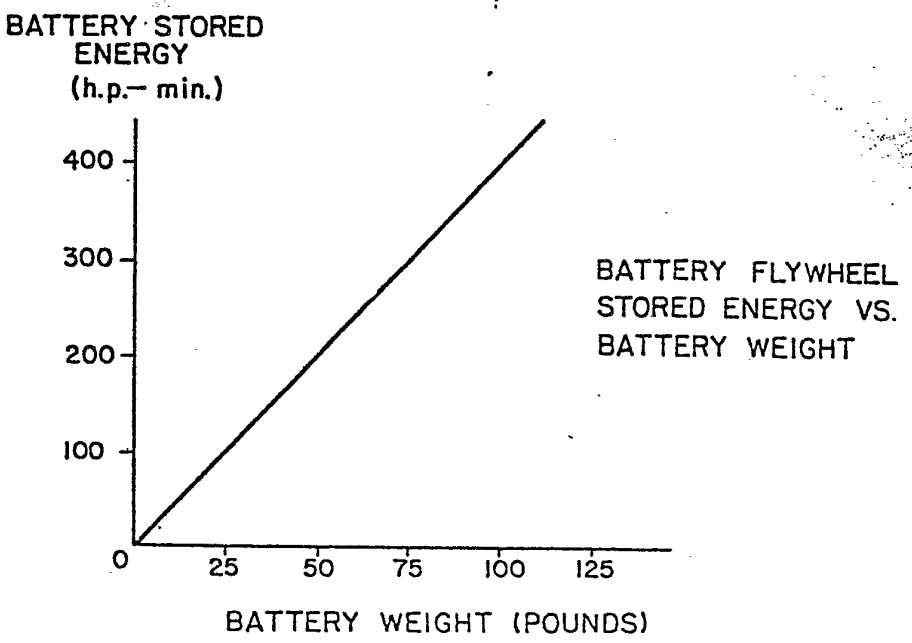
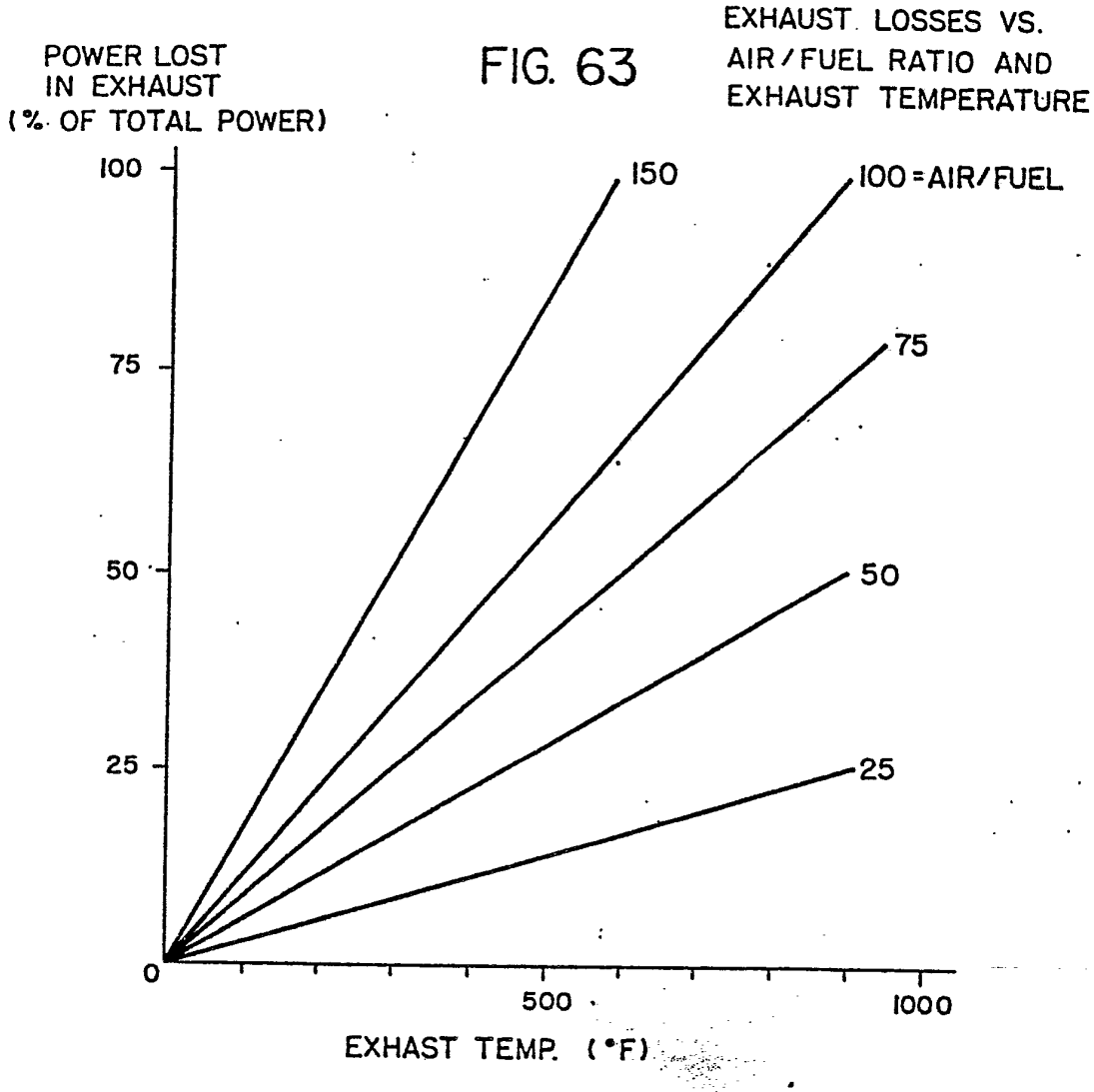
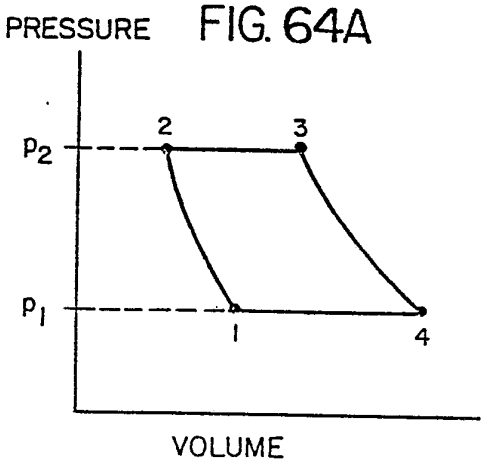


FIG. 62

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WIPO

FIG. 63





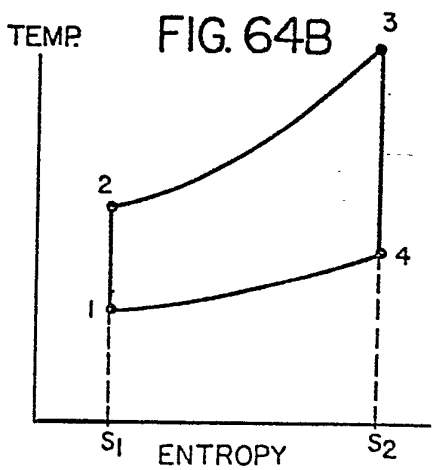


FIG. 64

BUREAU

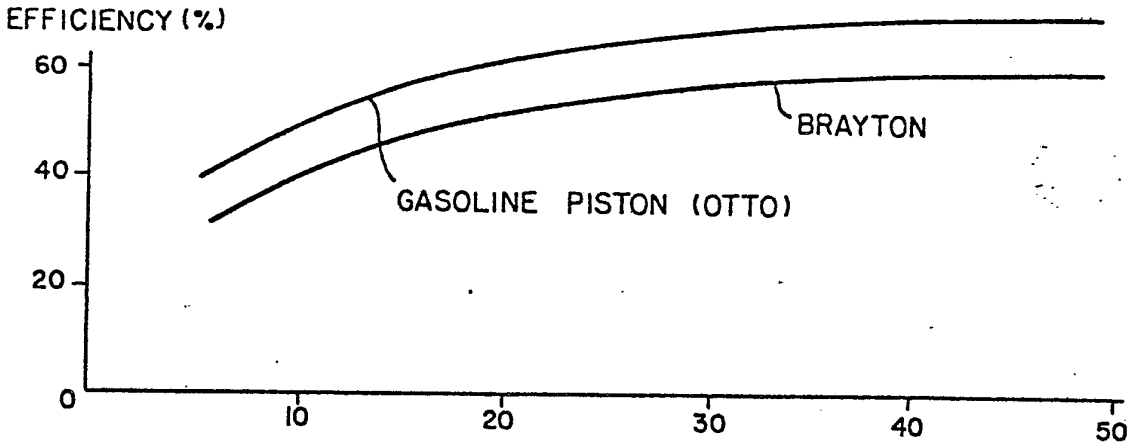


FIG. 65

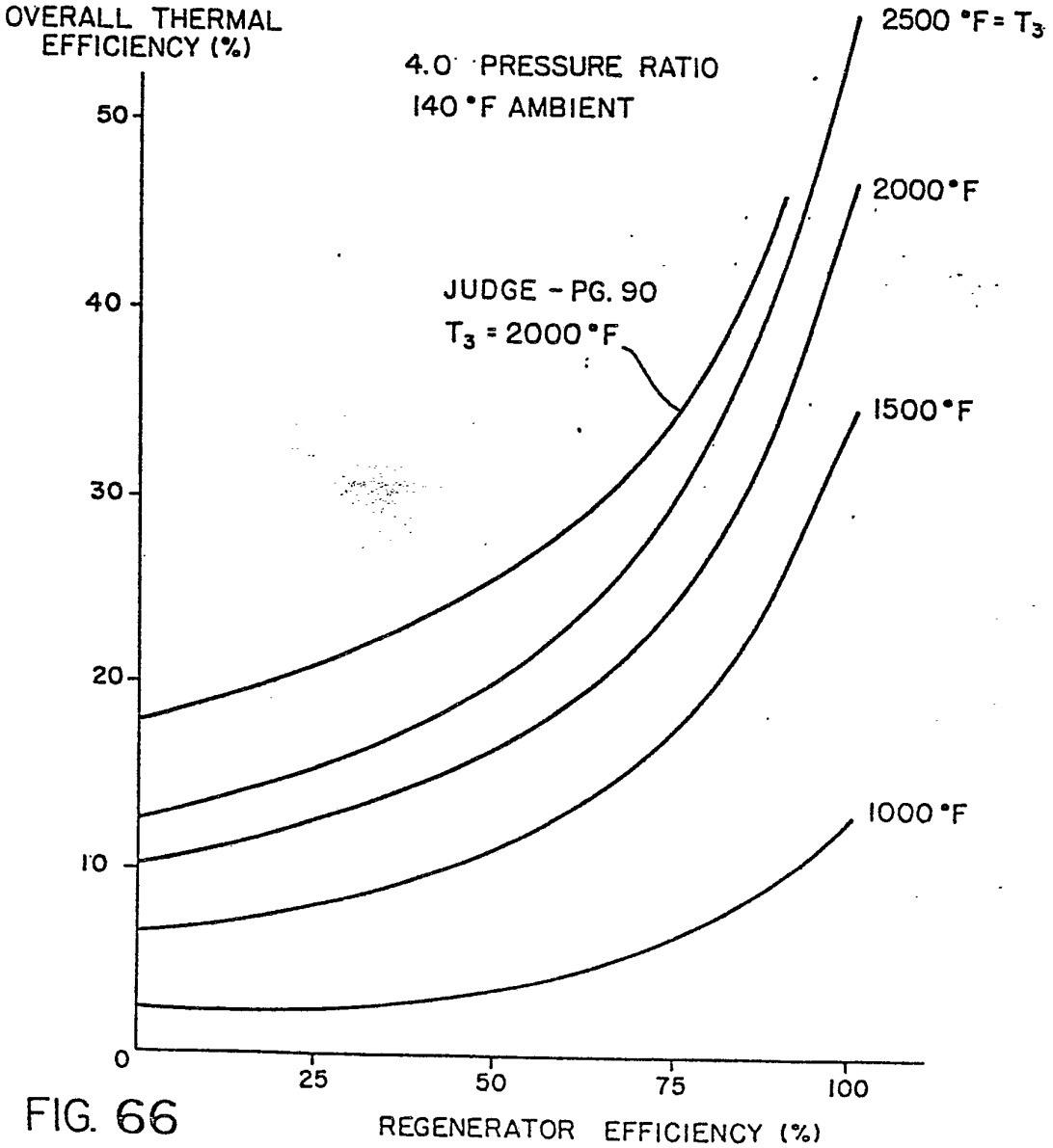


FIG. 66

BUREAU OMPI

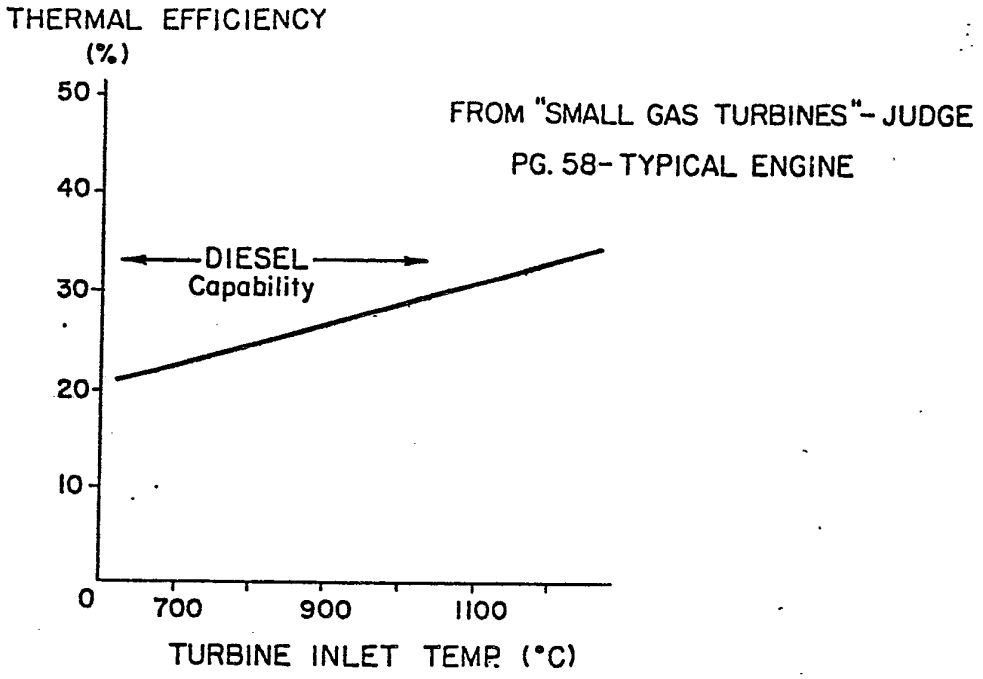


FIG. 67

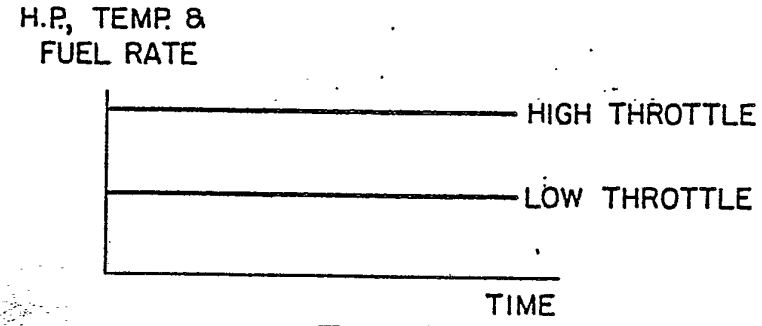


FIG. 68

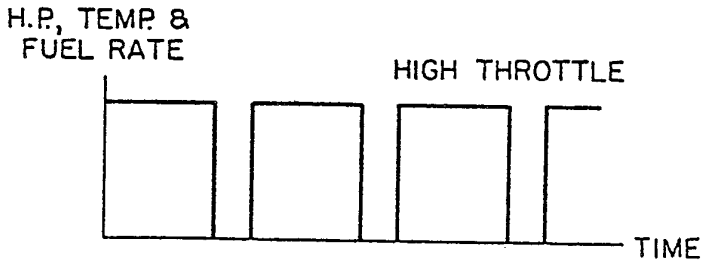


FIG. 69A

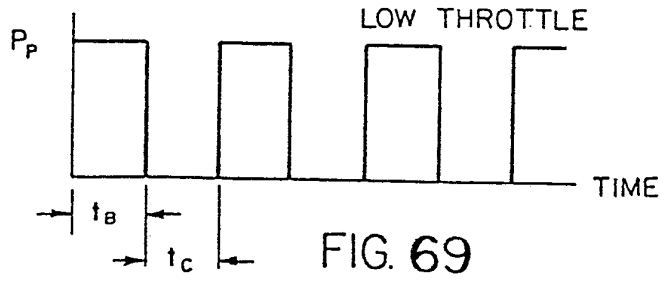


FIG. 69B

FIG. 69

BUREAU
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WIPO

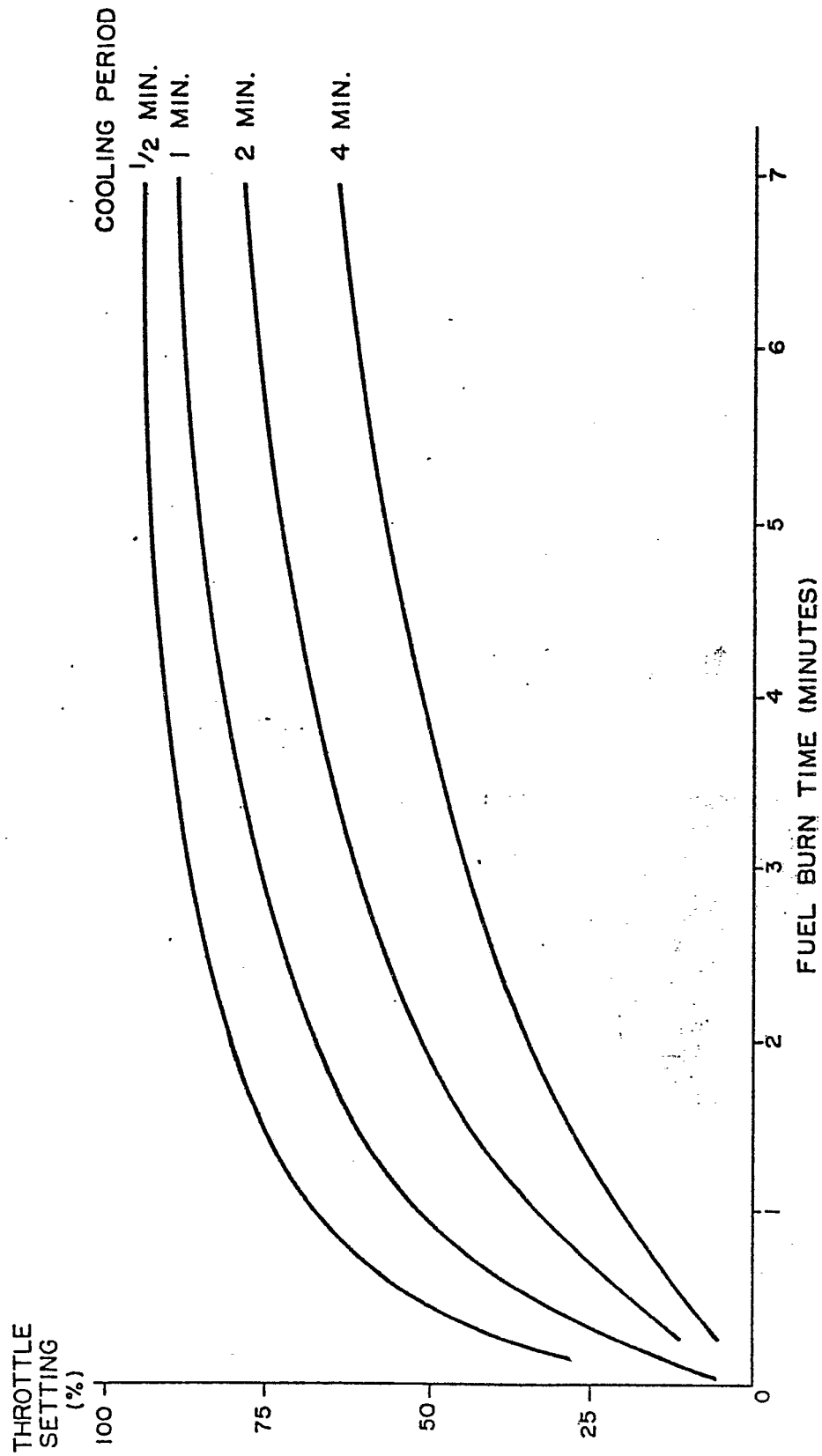


FIG. 70

BUREAU OMPI WIPO

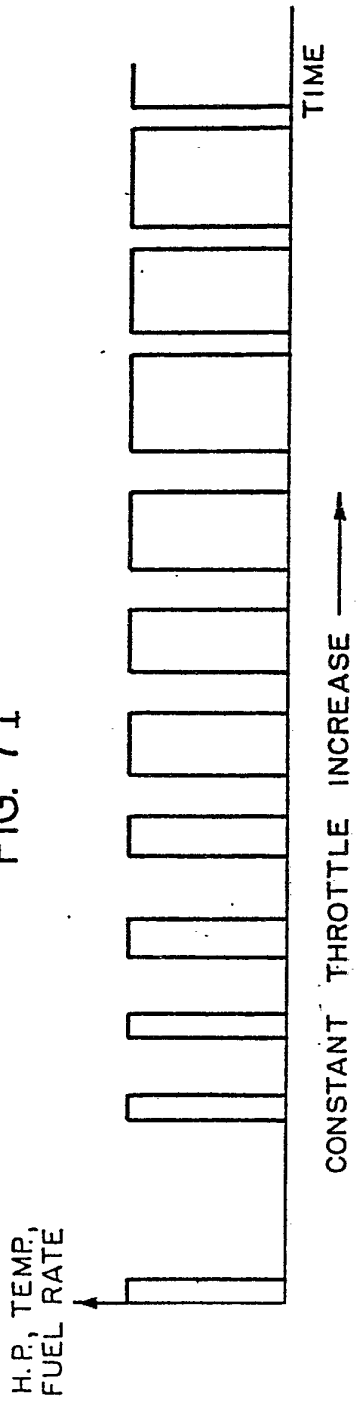


FIG. 71

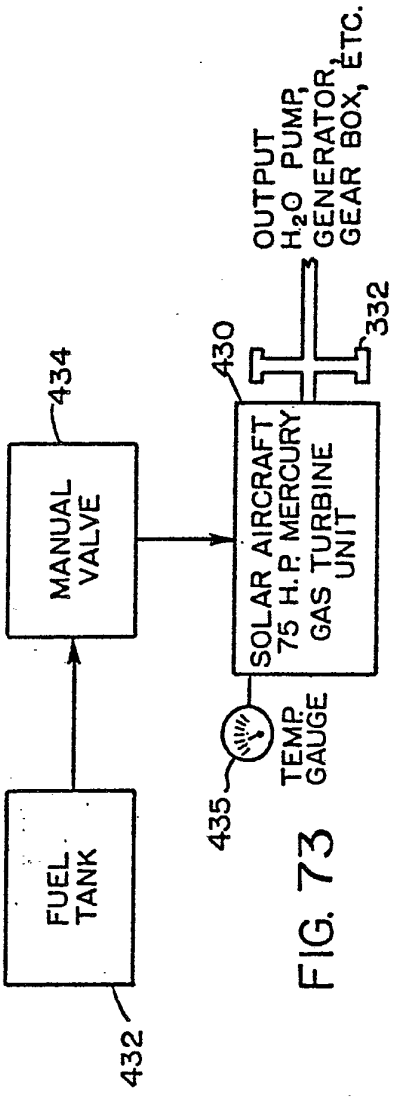


FIG. 73

BUREAU OMPI WIPO

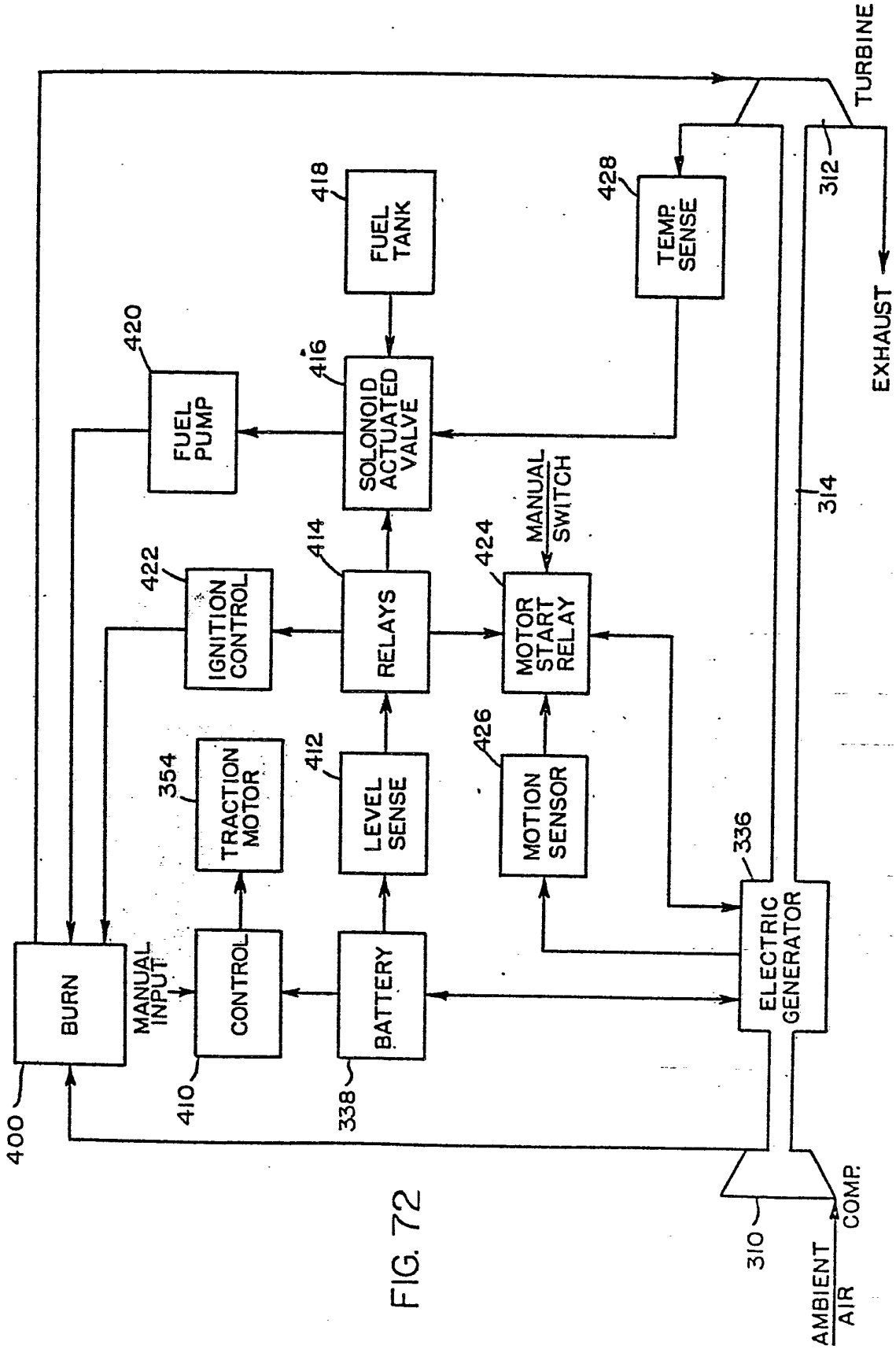


FIG. 72

BUREAU OMPI WIPO

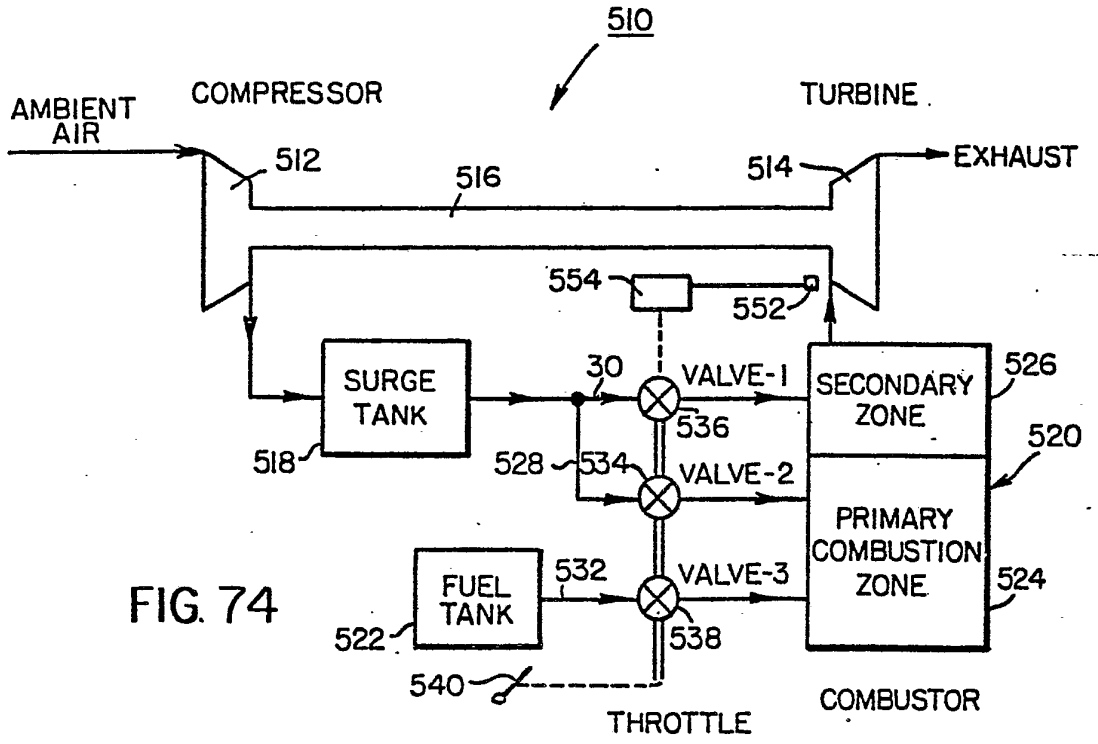


FIG. 74

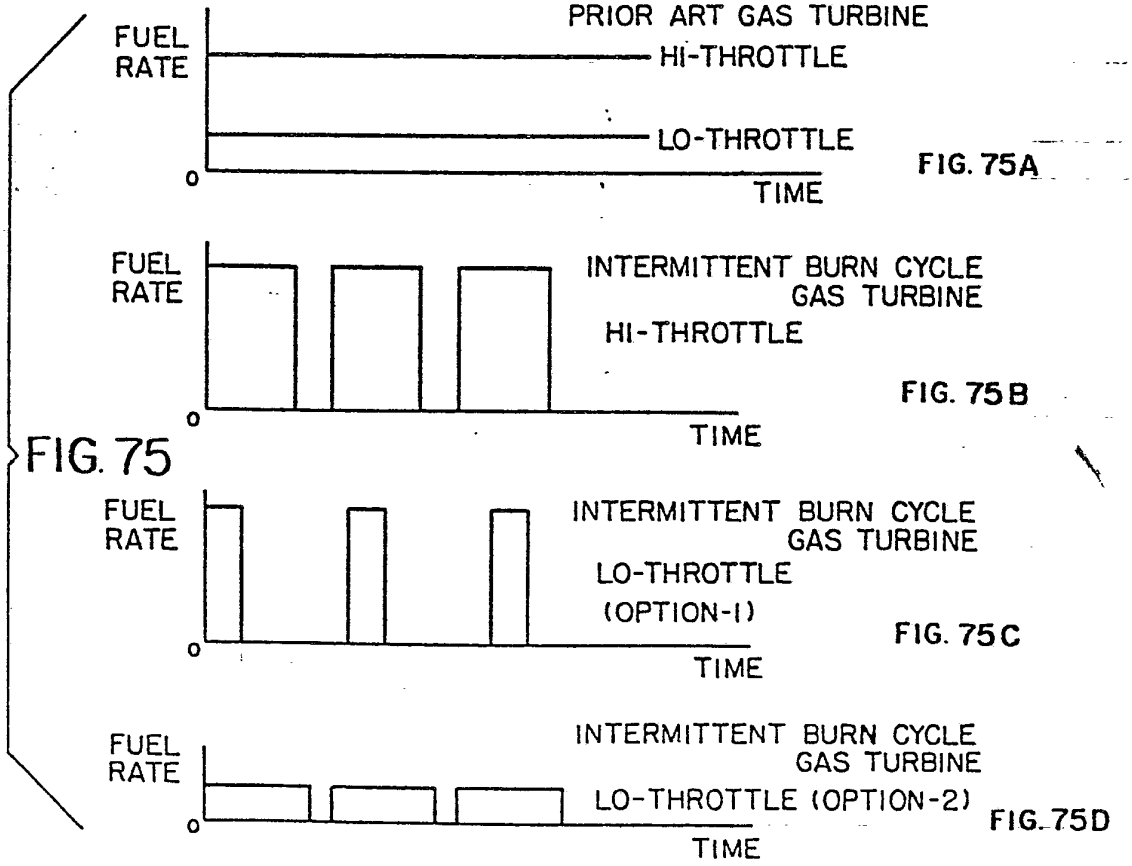


FIG. 75A

FIG. 75B

FIG. 75

FIG. 75C

FIG. 75D

BUREAU

FIG. 76

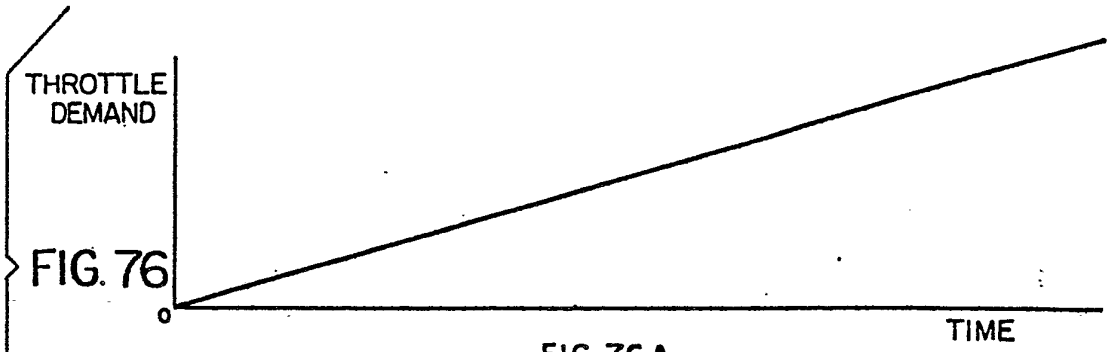


FIG. 76A

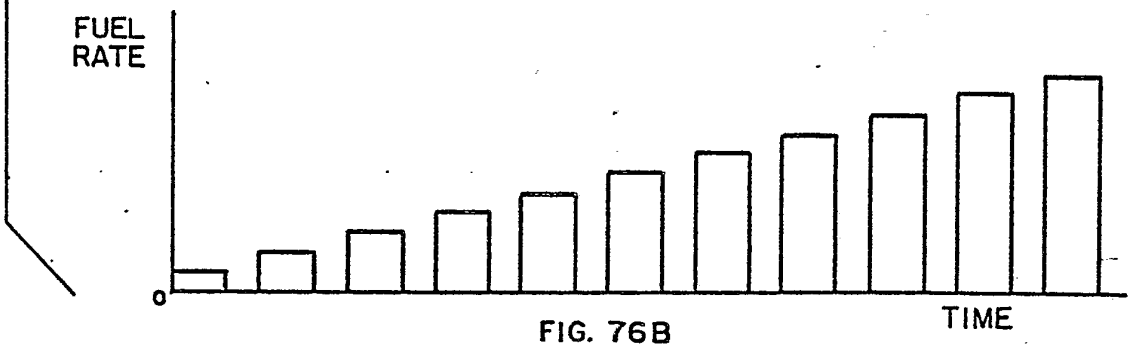


FIG. 76B

FIG. 77

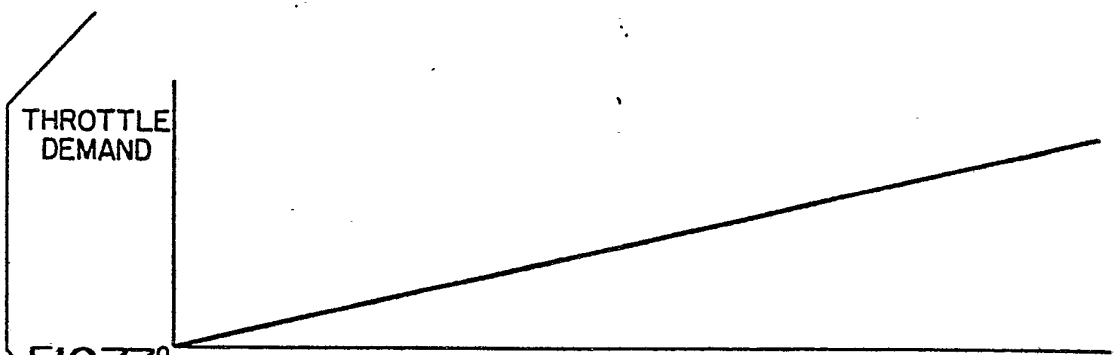


FIG. 77A

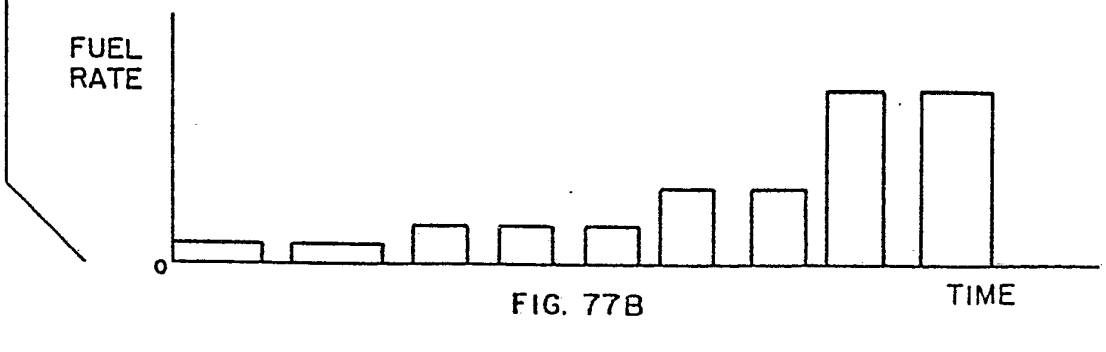
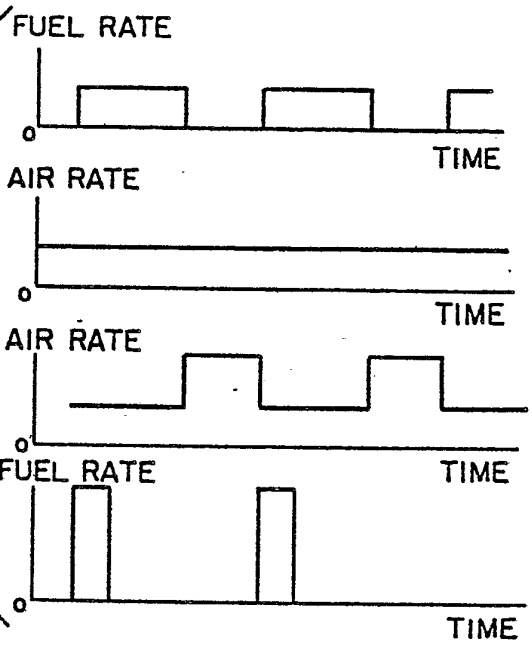


FIG. 77B

BUREAU OMPI WIPO

FIG. 78



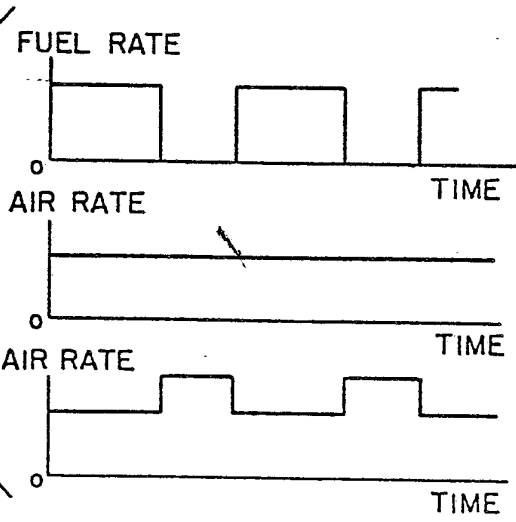
FUEL OPTION 1 FIG. 78A

AIR OPTION 1 FIG. 78B

AIR OPTION 2 FIG. 78C

FUEL OPTION 2 FIG. 78D

FIG. 79



FUEL OPTION 1 FIG. 79A

AIR OPTION 1 FIG. 79B

AIR OPTION 2 FIG. 79C

BUREAU

FIG. 80

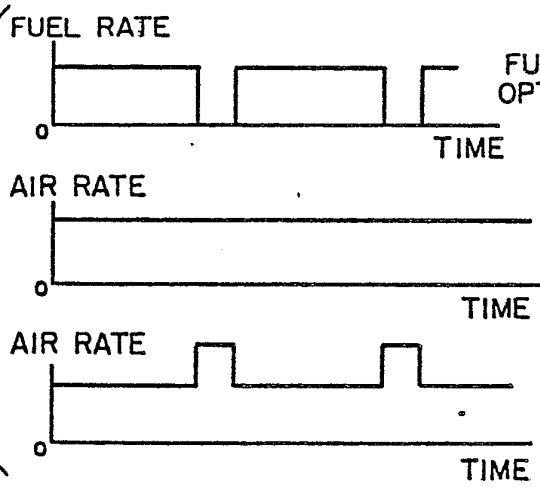
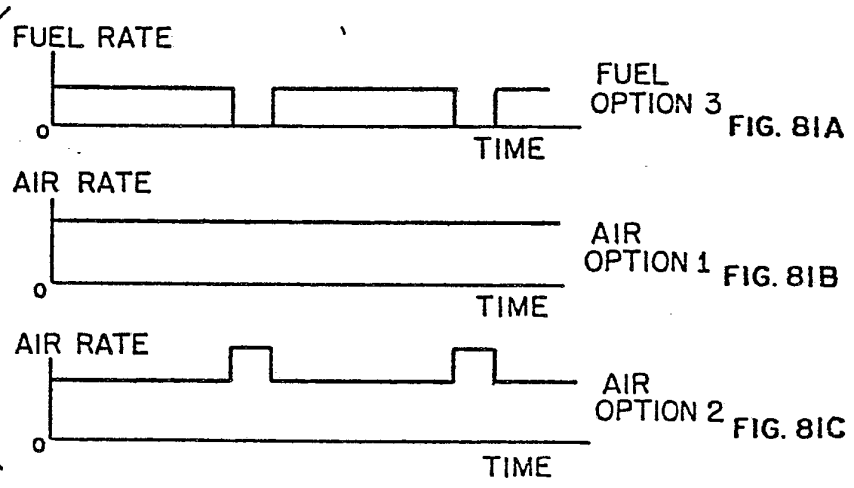


FIG. 81



BUREAU OMPI

FIG. 82

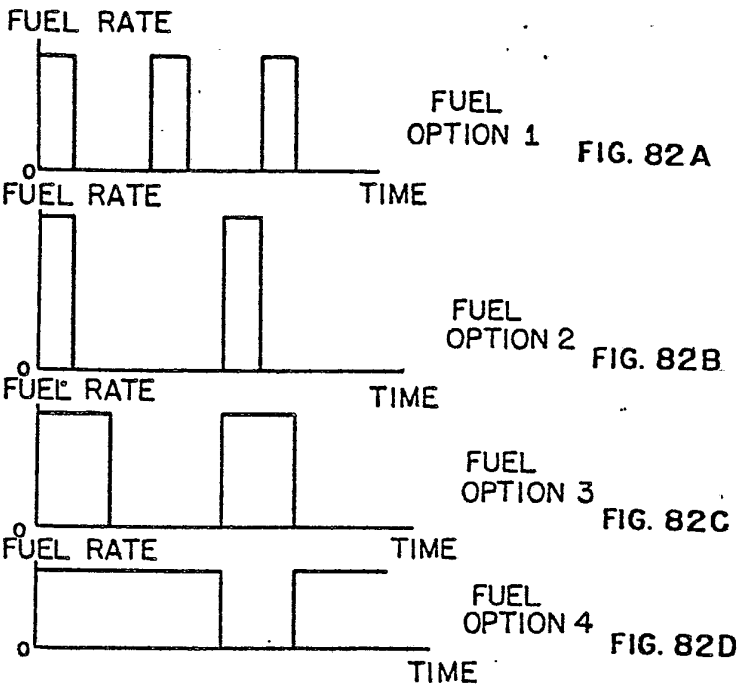
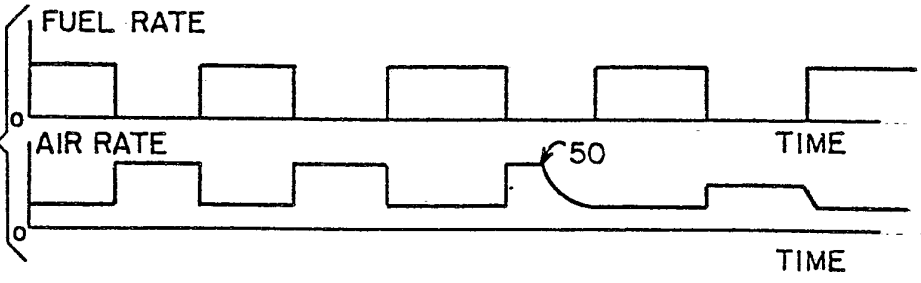


FIG.83



BUREAU OMPI WIPO

INTERNATIONAL SEARCH REPORT

WO 80/00170

International Application No PCT/US79/00318

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) 3

According to International Patent Classification (IPC) or to both National Classification and IPC

INTL CL F02C 5/00; F01K 25/00; F04C 29/00
US. CL. 60/39.76; 123/202; 123/204; 418/152; 60/650; 60/682

II. FIELDS SEARCHED

Minimum Documentation Searched 4

Classification System	Classification Symbols
US	60/39.63; 39.76; 39.78; 650; 682 123/202; 204; 418/152

Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched 5

III. DOCUMENTS CONSIDERED TO BE RELEVANT 14

Category *	Citation of Document, 16 with indication, where appropriate, of the relevant passages 17	Relevant to Claim No. 18
X	US,A 3057157, Published 9 October 1962 Close	1-6, 8-15, 18, 25-28, 30-34 36-67, 70-82
X	US,A 980801, Published 3 January 1911 Kraus	4-6, 18, 38, 57, 81
X	US,A,2872786, Published 10 February 1959 Ergenc	7, 16, 17, 31 60, 69
X	US,A 3293850, Published 27 December 1966 Morrison	7, 16, 17, 31, 60 69
X	US,A 3779216, Published 18 December 1973 Britt	12, 14, 15, 49/ 12
X	US,A 3191852, Published 29 June 1965 Kaatz et al	13, 18, 28
X	US,A 3672164, Published 27 June 1972 Pieper	19-24, 47, 48 49/19, 50/3, 50/8, 50/9, 51 70-75
X	US,A 3501913, Published, 24 March 1970	19-24, 47, 48

* Special categories of cited documents: 15

"A" document defining the general state of the art

"E" earlier document but published on or after the international filing date

"L" document cited for special reason other than those referred to in the other categories

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

"P" document published prior to the international filing date but on or after the priority date claimed

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

"X" document of particular relevance

IV. CERTIFICATION

Date of the Actual Completion of the International Search 2	Date of Mailing of this International Search Report 2
28 September 1979	16 OCT 1979
International Searching Authority 1	**Signature of Authorized Officer 20**
ISA/US	Michael Koczo Michael Koczo

Form PCT/ISA/210 (second sheet) (October 1977)

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

	Brille	49/19, 50/3, 50/8, 50/9, 51, 70-75
X	GB,A 947822, Published 29 January 1964 Birmann	19-24, 47, 48 49/19, 50/3, 50/8, 50/9 51, 70-75
X	AT,B 140501, Published 11 February 1935 Pistl et al	29,35
X	US,A 3228196, Published 11 January 1966 Paulsen	31,60,68,69

V. ☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE [10]

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1. ☐ Claim numbers _____, because they relate to subject matter [12] not required to be searched by this Authority, namely:

2. ☐ Claim numbers _____, because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out [13], specifically:

VI. ☐ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING [11]

This International Searching Authority found multiple inventions in this international application as follows:

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.

2. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:

3. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers:

Remark on Protest

☐ The additional search fees were accompanied by applicant's protest.

☐ No protest accompanied the payment of additional search fees.

FURTHER INFORMATION CONTINUED FROM THE SECOND SHEET

X	US, A, 3724427, published 3 April 1973 SAUDER	31,61
X	US, A, 3863610, published 4 February 1975 SPINNETT	35
X	US, A, 2461757, published 15 February 1949 MOORES	36, 80
X	US, A, 4018191, published 19 April 1977 Lloyd	39, 46

V. ☐ OBSERVATIONS WHERE CERTAIN CLAIMS WERE FOUND UNSEARCHABLE [10]

This international search report has not been established in respect of certain claims under Article 17(2) (a) for the following reasons:

1. ☐ Claim numbers _____, because they relate to subject matter [12] not required to be searched by this Authority, namely:

2. ☐ Claim numbers _____, because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out [13], specifically:

VI. ☐ OBSERVATIONS WHERE UNITY OF INVENTION IS LACKING [11]

This International Searching Authority found multiple inventions in this international application as follows:

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims of the international application.

2. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims of the international application for which fees were paid, specifically claims:

3. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claim numbers:

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☐ The additional search fees were accompanied by applicant's protest.

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