United States Patent [19]

Engdahl

[11] **3,774,397**

[45] **Nov. 27, 1973**

[54] **HEAT ENGINE**

[75] Inventor: **Richard E. Engdahl,** Danbury, Conn.

[73] Assignee: **Energy Research Corporation,** Bethel, Conn.

[22] Filed: **Aug. 4, 1971**

[21] Appl. No.: **168,866**

[52] **U.S. Cl.** .... **60/107,** 60/1, 137/59, 277/96 A, 277/27, 418/153, 418/178, 418/266

[51] **Int. Cl.** **F01k 13/02,** F01c 17/00

[58] **Field of Search** .... 418/25, 178, 265; 92/169–171; 308/160, 241; 277/96 A, 27; 60/36, 108, 106, 39, 40, 105, 107

[56] **References Cited**

UNITED STATES PATENTS

2,588,342	3/1952	Bidwell	418/158 X
3,386,648	6/1968	Van Rossen	418/93 X
745,755	12/1903	Allen	418/25
1,831,411	11/1931	Dietz	92/169
2,721,105	10/1955	Schaefer	308/160
3,057,157	10/1962	Close	60/108
3,141,677	7/1964	Williams	277/27 X
3,147,013	9/1964	Tracy	277/96 A
3,155,313	11/1964	Bentele	418/178
3,169,375	2/1965	Velthuis	60/108
3,388,553	6/1968	Anderson	60/107 X
3,591,243	7/1971	Knoppenberg et al.	308/241
3,545,207	12/1970	Barber et al.	60/106

FOREIGN PATENTS OR APPLICATIONS

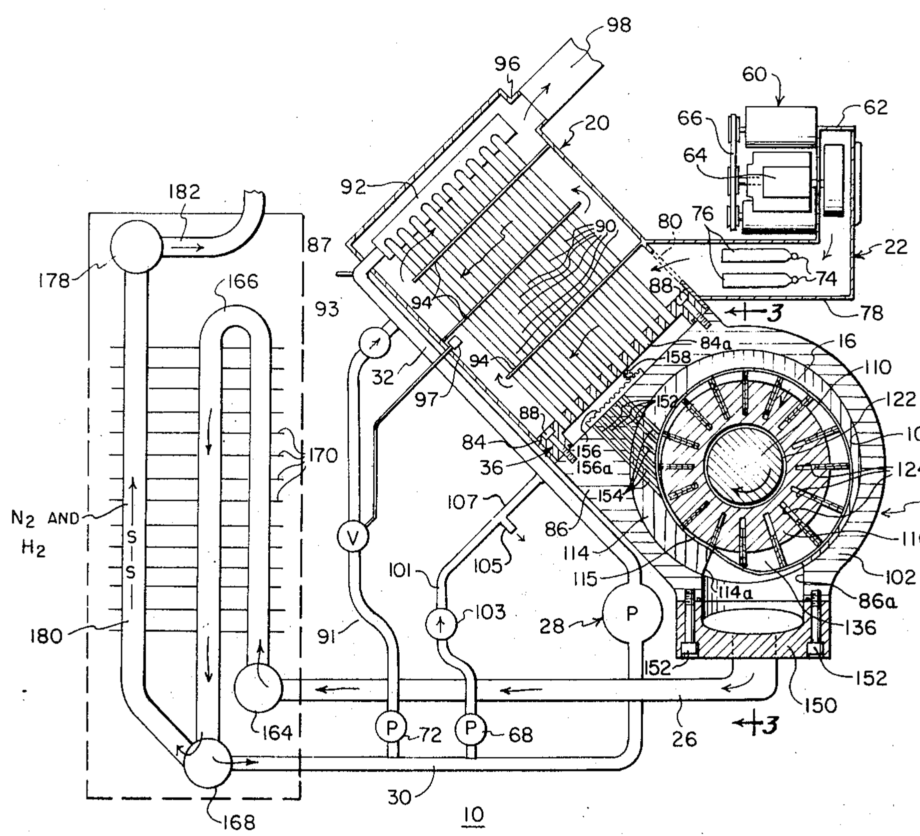
241,172	3/1969	U.S.S.R.	277/27
58	1/1874	Great Britain	418/265
26,529	8/1908	Great Britain	418/265
945,265	12/1963	Great Britain	277/96 A

Primary Examiner—Martin P. Schwadron
Assistant Examiner—Allen M. Ostrager
Attorney—Richard D. Mason et al.

[57] **ABSTRACT**

A substantially pollution-free heat engine employing steam as the working fluid and operable on a cycle approaching a Rankine cycle. The engine includes a lightweight, small-size, highly efficient, boiler expander and condenser system which utilizes advanced material technology in order to achieve maximum system operating temperatures of approximately 1,200°F. and operating pressures up to 3,500 psi. The engine is especially suitable for use in motor vehicles and is capable of fast start-ups so as to produce 65 percent of maximum power within 45 seconds or less. Moreover, the engine provides an idle mode of operation for driving vehicle accessories and the like and in addition is responsive to produce rapid power changes needed for acceleration and heavy pulls on steep grades.

17 Claims, 7 Drawing Figures



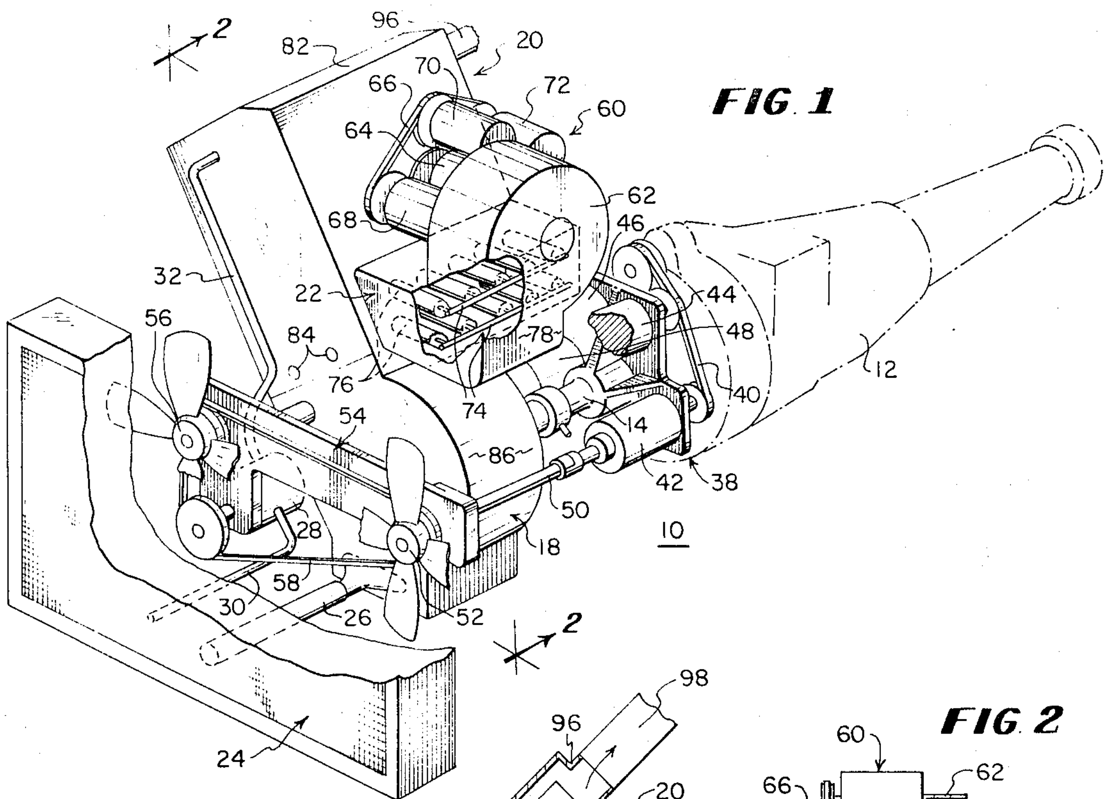


FIG. 1

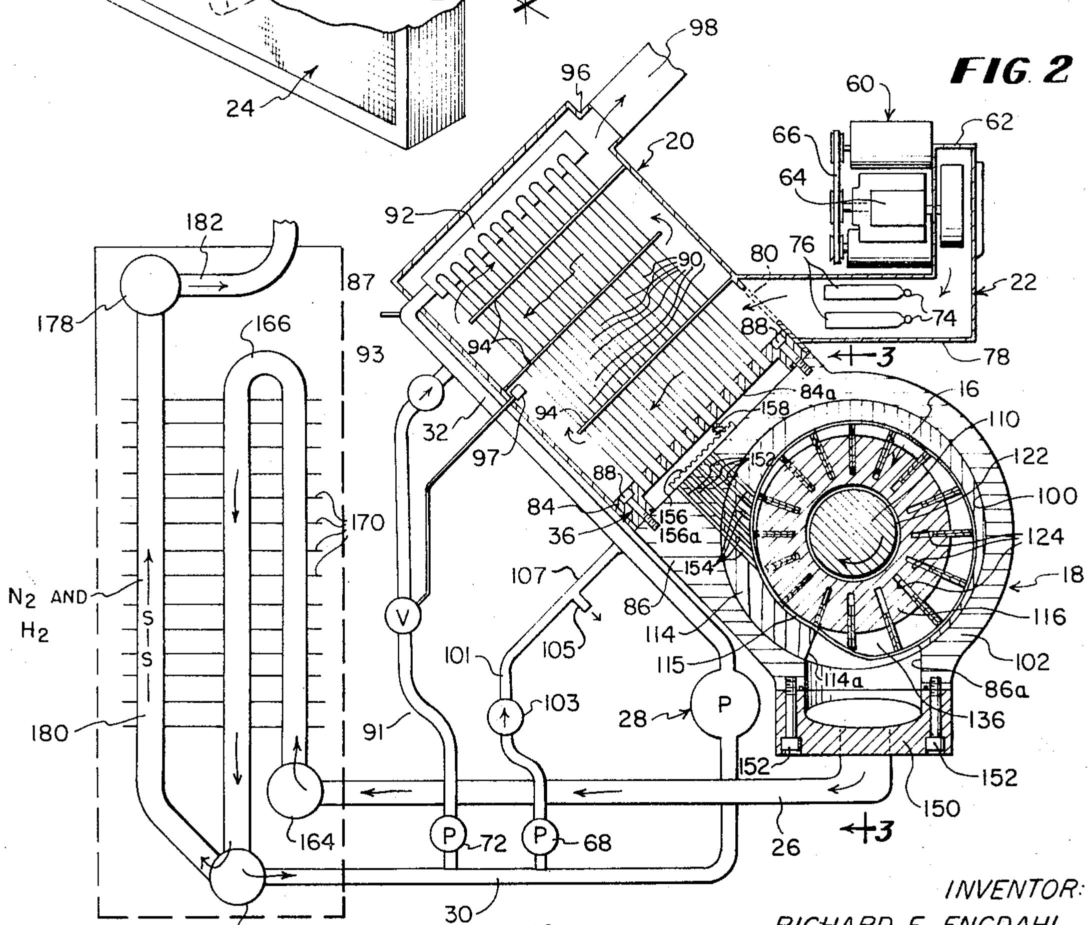


FIG. 2

INVENTOR:
RICHARD E. ENGDAHL

BY Mason, Kolehmainen, Rathburn & Wyss
ATTORNEYS

Patented Nov. 27, 1973 3,774,397

4 Sheets-Sheet 2

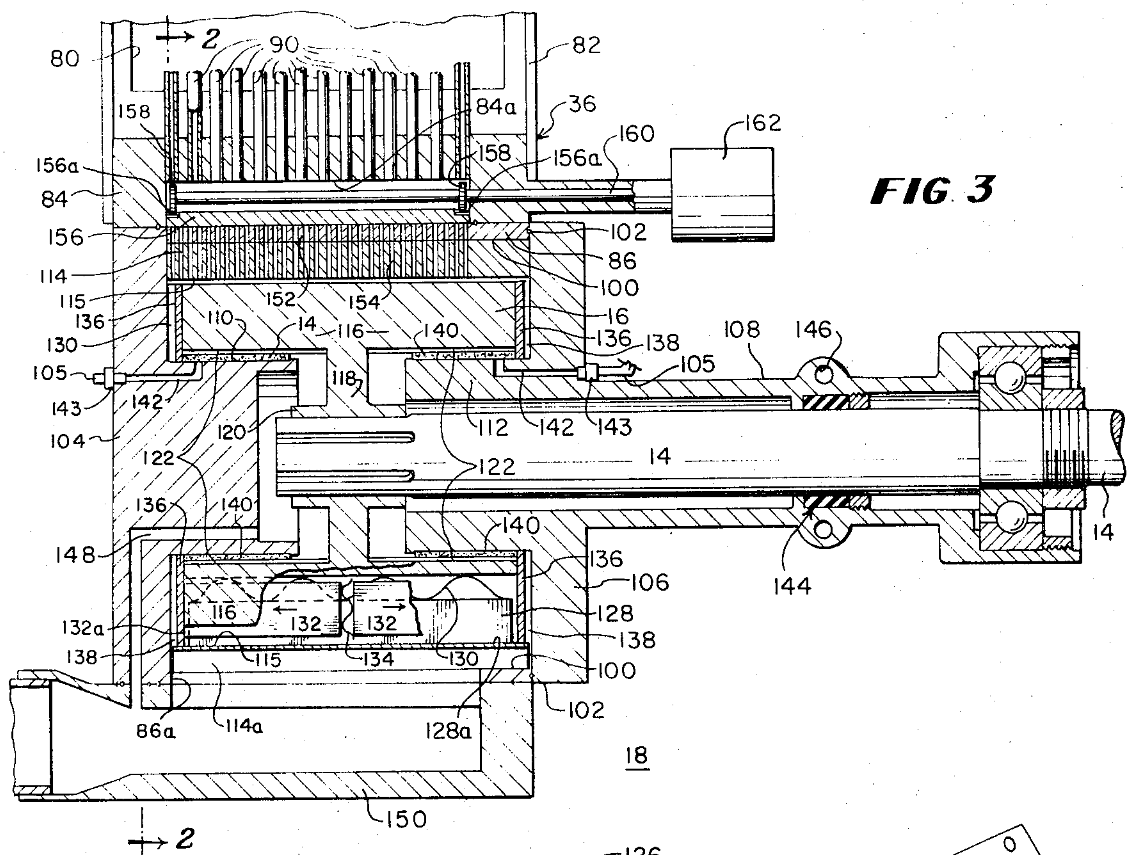


FIG. 3

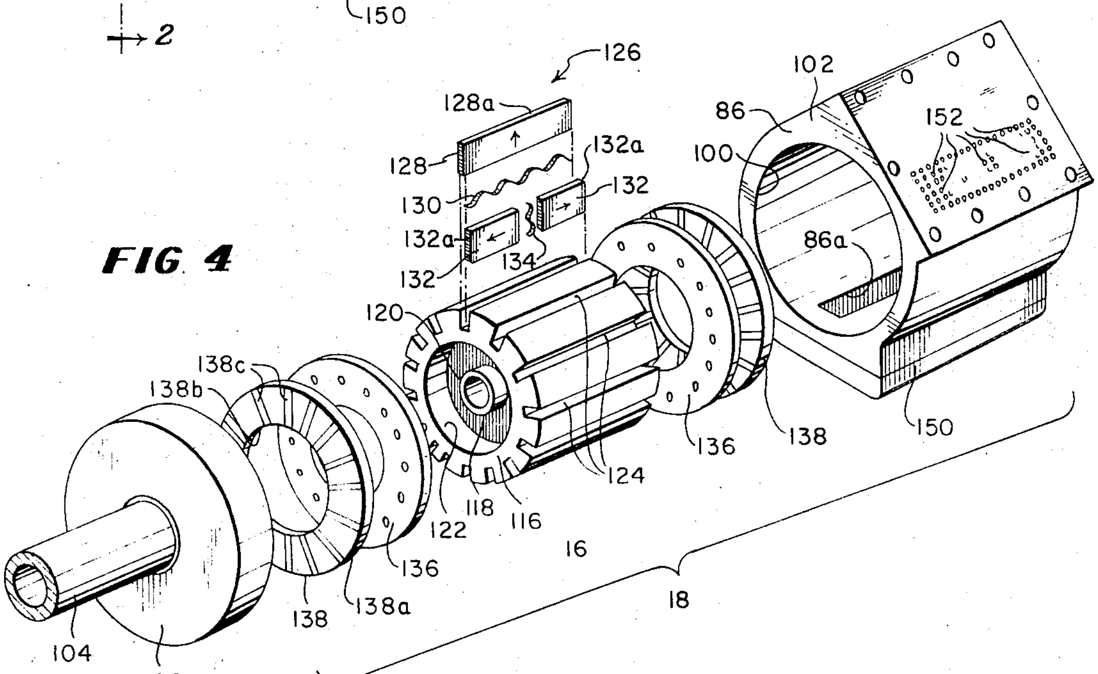


FIG. 4

INVENTOR:
RICHARD E. ENGDAHL

BY Mason, Kolehmainen Rathburn & Wyss
ATTORNEYS

FIG. 5

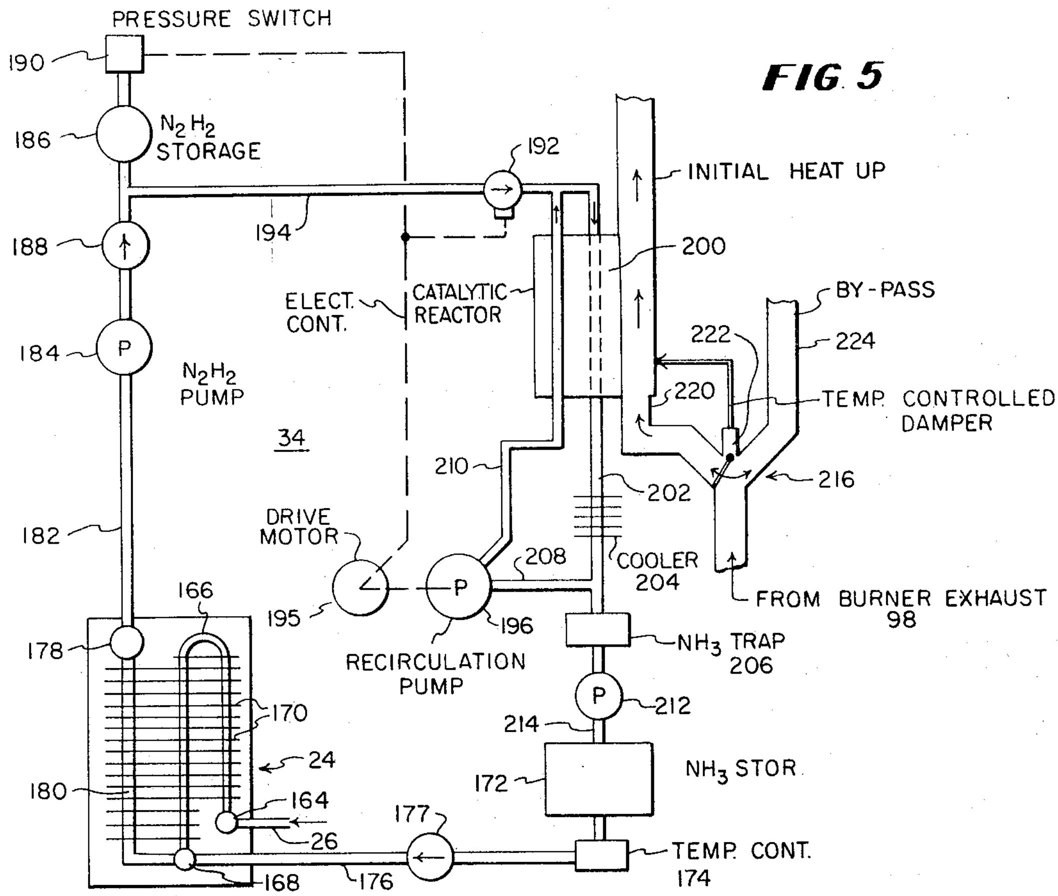
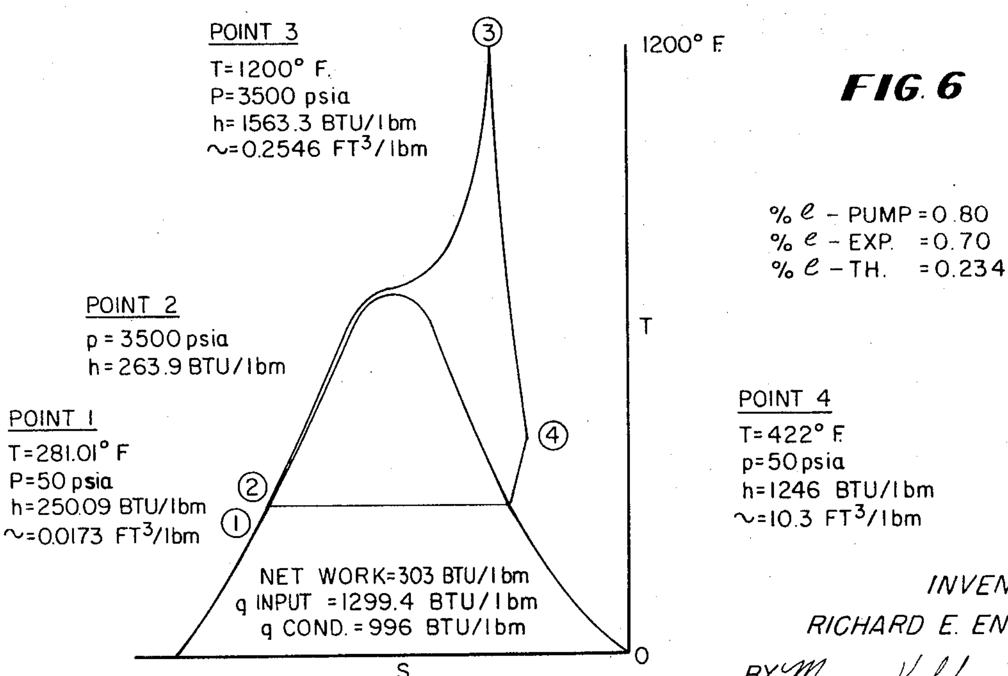


FIG. 6

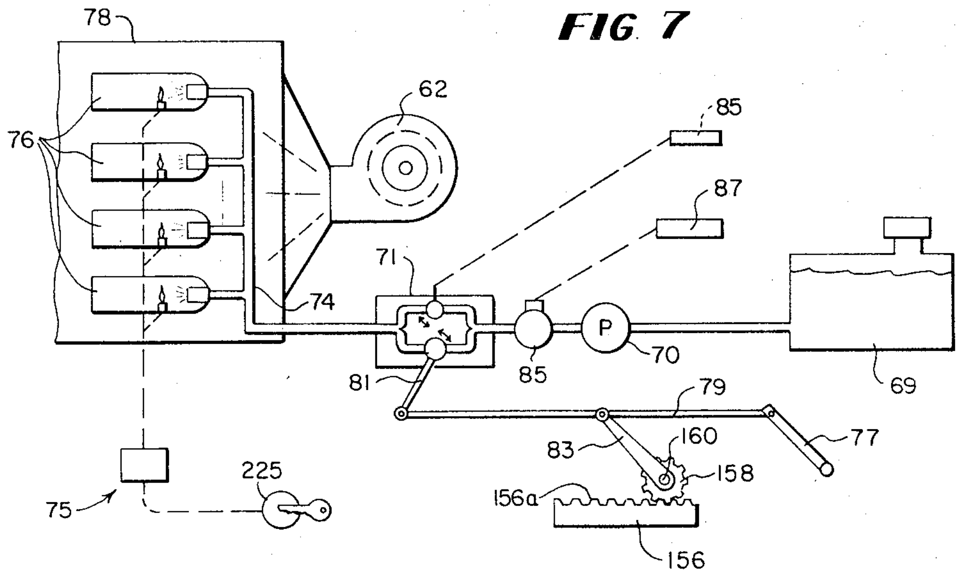


INVENTOR:
RICHARD E. ENGDAHL
BY Mason, Kolehmainen Rathburn & Wyss
ATTORNEYS

Patented Nov. 27, 1973 3,774,397

4 Sheets-Sheet 4

FIG. 7



INVENTOR:
RICHARD E. ENGDAHL

BY Mason, Kolehmainen
Rathburn & Wyss
ATTORNEYS

HEAT ENGINE

The present invention relates to a new and improved low pollution heat engine and, more particularly, relates to an engine utilizing superheated steam as the working fluid at relatively high temperatures and pressures in an operating cycle approximating the Rankine cycle.

The novel heat engine of the present invention employs a high temperature, high pressure, small-size, highly efficient fluid expander which utilizes advanced material technology in overcoming difficult frictional sliding and pressure sealing problems inherent in such high temperature, high pressure operations. Because the working fluid used condenses to water, a freezing problem in cold weather must be overcome and accordingly a unique use of ammonia gas as an antifreeze is provided. The boiler of the engine is effective to provide extremely high heat release rates, has a low mass, and is of minimal size in order to provide a rapid start-up capability. In addition, the heat engine of the invention employs a burner system which provides more complete combustion and produces minimal amounts of pollutants in the exhaust. Additionally, the engine includes an ammonia synthesis system for recombining any hydrogen and nitrogen gas which becomes chemically dissociated from the ammonia antifreeze during the operation of the engine because of the high temperatures and pressures of the working fluid.

It is therefore an object of the present invention to provide a new and improved substantially pollution-free heat engine and, more particularly, it is an object of the present invention to provide a new and improved heat engine of the character described which is extremely well suited for powering motor vehicles, and the like, and which is capable of combination with transmisssion and wheel drive assemblies presently available.

Another object of the present invention is to provide a new and improved substantially pollution-free heat engine of the character described which operates on a thermal cycle approximating the theoretical Rankine cycle.

Another object of the present invention is to provide a new and improved heat engine of the character described which utilizes superheated steam as a working fluid at working temperatures of approximately 1,200°F. and pressures of approximately 3,500 psia.

Still another object of the present invention is to provide a new and improved heat engine of the character described which employs a novel positive displacement expander capable of operating at high pressure ratios, with a high volume ratio of expansion between inlet and outlet.

Another object of the present invention is to provide a new and improved positive displacement expander of the character described which provides an extremely high volume expansion ratio, yet in a single stage system and which provides a substantially continuous flow system having high efficiency.

Another object of the invention is to provide a new and improved positive displacement fluid expander of the character described having novel control means for regulating the power output.

Another object of the present invention is to provide a new and improved heat engine of the character described having novel boiler means which is capable of generating supercritical steam at high temperature and pressure in a rapid continuous process and with a rapid start-up capability.

Another object of the present invention is to provide a new and improved heat engine having boiler means that is small in size, light in weight, and extremely efficient in operation.

Another object of the present invention is to provide a new and improved heat engine having a new and improved condenser system providing a high efficiency of heat transfer, yet being relatively small and compact in size and weight.

Another object of the present invention is to provide a new and improved low pollution, Rankine cycle heat engine of the character described which is extremely well suited for supplying power for motor vehicles, and the like.

Still another object of the present invention is to provide a new and improved heat engine of the character described utilizing superheated steam as a working fluid, and yet being relatively easy to control in terms of power output over a wide variety of speed and torque ranges.

Another object of the present invention is to provide a new and improved engine of the character described which is especially well suited to serve as a direct replacement or alternate for presently available internal combustion engines used in motor vehicles.

Still another object of the present invention is to provide a new and improved engine of the character described which can be readily substituted for presently available internal combustion engines and which engine is comparable or smaller in size, lighter in weight, and equal or lower in cost, both from an original cost standpoint and from an operating cost standpoint.

Another object of the present invention is to provide a new and improved heat engine of the character described which operates in a safe, reliable manner, one which is economical in operation, and which meets acceleration and cruise performance requirements for normal stop and go driving.

Another object of the present invention is to provide a new and improved heat engine of the character described which produces a relatively low level of pollutants which are discharged from the exhaust system into the atmosphere.

The foregoing and other objects and advantages of the present invention are accomplished in an illustrated embodiment comprising a new and improved heat engine utilizing supercritical, high temperature, high pressure steam as the working fluid and operating on a cycle approximating the Rankine cycle. The heat engine includes a main boiler feed pump; a compact, lightweight, high capacity boiler system; a high heat capacity burner system firing the boiler; and a new and improved positive displacement, high volume ratio expander driving a rotary output shaft. Spent working fluid leaving the expander flows into a condenser, and a new and improved antifreeze system using ammonia is provided including an ammonia synthesis unit.

For a better understanding of the present invention, reference should be had to the following detailed description taken in conjunction with the drawings, in which:

FIG. 1 is a perspective view of a new and improved heat engine constructed in accordance with the features of the present invention and shown as it might be installed to power the drive shaft of a motor vehicle, and the like, through presently available vehicle transmission means;

FIG. 2 is the transverse cross-sectional view of the main body portion of the heat engine of FIG. 1 taken substantially along line 2—2 thereof illustrating the condenser of the engine in schematic form;

FIG. 3 is a longitudinal cross-sectional view of the heat engine taken substantially along line 3—3 of FIG. 2;

FIG. 4 is an exploded perspective view of the positive displacement expander of the engine;

FIG. 5 is a schematic diagram illustrating the freeze protection and ammonia synthesis system of the engine;

FIG. 6 is a temperature-entropy diagram of the engine, and

FIG. 7 is a schematic diagram of the power control system of the heat engine.

Referring now, more particularly to the drawings, in FIGS. 1 and 2 is illustrated a new and improved heat engine constructed in accordance with the present invention and referred to generally by the reference numeral **10.** The engine **10** is designed to use high pressure, high temperature, superheated steam as the working fluid and operates on a cycle approximating the Rankine cycle as depicted in FIG. **6,** wherein entropy is the abscissa of the graph and temperature is the ordinant. At the low pressure point in the working cycle, as illustrated at point **1,** the water is at a temperature of about 281°F., a pressure of about 50 psia, and has an enthalpy of heat value of 250.09 BTU's per pound. At this point, the working fluid has a specific volume of 0.0173 cu. ft. per pound. The fluid is passed through a boiler feed-water pump wherein the pressure is increased from 50 to 3,500 psia, and a slight increase in the enthalpy to 263.9 BTU's per pound is produced at point **2.** From point **2,** heat is added in the boiler to the working fluid to produce superheated steam and during this process, the fluid temperature is increased to a maximum at point **3** of approximately 1,200°F. with the pressure remaining about the same and with a significantly large increase in the enthalpy up to 1,563.3 BTU's per pound. The specific volume of the fluid at point **3** is also increased 0.2546 cu. ft. per pound. The working fluid at point **3** comprising superheated steam is then expanded to do work in the positive displacement expander of the heat engine and a significantly high expansion ratio of approximately 70 to 1 (pressure ratio) is provided. The temperature at the end of the expansion phase (point **4**) is reduced to approximately 422°F. and the pressure is down to approximately 50 psia with a significant decrease in enthalpy down to 1,246 BTU's per pound representing the work done during expansion. During the expansion phase, a significant increase in the specific volume up to approximately 10.3 cu. ft. per pound is experienced representing an expansion volume of roughly 40 to 1.

In the condensing phase of the cycle, the fluid returns from the conditions at point **4** to the starting condition at point **1** and in the condensing process heat is taken from the working fluid at a rate or approximately 996 BTU's per pound in comparison to the heat input to the fluid in the boiler phase of approximately 1,299.4 BTU's per pound. The net work output of the heat engine is approximately 303 BTU's per pound and the thermal efficiency is based on an assumed feed-water pump efficiency of approximately 80 percent and an expander efficiency of roughly 70 percent. The resultant thermal efficiency of the engine is approximately 23.4 percent, which is extremely good for a heat engine of this type. It should also be noted that with the advanced material technology applied, working fluid maximum temperatures of 1,200°F. and pressures of 3,500 psia are present and are handled by the engine components without extreme difficulty. These high temperatures and pressures of the working fluid, together with the high volume ratio of expansion in the expander (approximately 40 to 1) and the favorable air-cooled condenser temperature differentials, provide a heat engine that is considerably smaller in size, lighter in weight, and more economical to operate than other types of motor vehicle steam powered engines heretofore proposed.

In accordance with the present invention, the heat engine **10,** as shown in FIG. **1,** is relatively compact in size and light in weight in order to provide suitable motive power for small automobiles and other powered vehicles. The engine is adapted to supply power at conventional speed and torque ranges through presently available transmissions, for example, as indicated generally at **12** by the dotted lines in FIG. **1.** The power output of the engine is delivered to the transmission **12** through a main output shaft **14** which is driven by a rotor assembly **16** (FIGS. **2** and **3**) of a positive displacement expander, generally indicated by the reference numeral **18.** Working fluid at high pressure and temperature is supplied to the positive displacement expander **18** from a boiler **20** mounted thereon, and the boiler is fired by a burner assembly generally indicated by the reference numeral **22.** After the working fluid is expanded and work output is obtained from the fluid in the positive displacement expander **18,** the fluid is passed into an air-cooled condensing unit **24** via a condensate return line **26.** After passing through the cooling coils of the condenser, the condensate is directed to a main feed-water pump **28** through a feed-water inlet line **30.** High pressure fluid is directed from the output side of the main feed-water pump into the boiler **20** through the feed-water supply conduit **32.**

In order to prevent the working fluid, when in condensate form (water), from freezing while the engine is not in operation during ambient temperature conditions below freezing, the heat engine **10** includes a freeze protection or anti-freeze system **34** shown in schematic diagram in FIG. **5.** The freeze protection or anti-freeze system includes means for injecting ammonia gas into the liquid condensate in response to ambient temperature conditions below freezing, and includes means for synthesizing ammonia from any hydrogen and nitrogen gas which may become chemically dissociated or cracked and separated from the working fluid of the heat engine because of the relatively high temperatures and pressures involved.

Because of the high expansion volume ratio provided in the positive displacement expander **18,** and even though relatively high inlet pressures are involved, the output shaft **14** of the engine is driven at speeds which are commensurate with shaft speeds of presently available internal combustion engines used for automobiles and other motor vehicles.

For the purpose of controlling the power output of the engine, a control system **36** (FIGS. **2** and **3**) is provided to regulate the flow of superheated steam from the boiler 20 into the positive displacement expander 18.

The engine arrangement shown in FIG. 1 is suitable for installation in a forward or rear engine compartment of a motor vehicle and, in the typical vehicular installation, a first accessory drive assembly 38 is driven from the output shaft 14 by a belt 40 which is trained around a number of pulleys to drive a generator or alternator 42, an air-conditioning unit 44, a hydraulic pump for power steering and power brakes 46, and an ammonia pump 48 used in the freeze protection system 34. As shown, the shaft of the generator or alternator 42 is connected through an extension shaft 50 to drive a first cooling fan 52 associated with the condenser 24. The cooling fan 52 is mounted on a second accessory drive assembly adjacent the forward end of the engine behind the condenser 24, and the second accessory drive assembly includes a second cooling fan 56 driven via a belt 58 which also drives the main feed-water pump 28 through a suitable drive pulley as shown.

In addition to the first and second accessory drive sections 38 and 54, the heat engine 10 also includes a third accessory drive assembly 60 which comprises a blower 62 for supplying combustion air to the burner system 22. As best shown in FIGS. 1 and 2, the blower 62 may be of the squirrel-cage type having an axial inlet opening and a discharge facing downwardly in communication with the outer end of the burner system 22. The blower 62 generates a relatively high volume of airflow at a suitable pressure (for example 20 inches of water) to flow through the burner and boiler, and is driven by an electric motor 64 having a pulley on the outer end of the shaft for driving other accessories through a belt 66. Thes accessories comprise a lubrication pump 68, a fuel pump 70, and a starting feed-water pump 72.

In accordance with the present invention, heat energy for the engine 10 is developed by the combustion of a suitable fuel, such as unleaded gasoline, kerosene, natural gas, or other gaseous or liquid hydrocarbons, such as butane or propane, which are delivered from suitable storage tanks 69 through a fuel pump 70 and a fuel control 71 to a fuel supply manifold 74, connected to the rearward ends of a plurality of jet engine type burners 76. An automatic burner ignition system 75 along with fuel control is provided for initiating and maintaining combustion in the burners 76. The combustion process is continuous, as in a turbo-jet engine, at variously selected rates from idle to maximum power until the engine is shut down. The fuel control 71 meters fuel to the burners 76 in accordance with the position of an engine throttle 77 (FIG. 6) which is connected to the fuel control through suitable linkage 79 and 81. The throttle also controls the position of a control plate 156 for controlling the flow of steam from the boiler 20 into the expander 18 as will be described hereinafter. The throttle linkage 79 is connected to rotate a shaft 160 through a link arm 83, and the shaft through a pinion 158 moves the control plate. In addition to manual control the fuel control 71 includes a bypass system to increase or decrease additional fuel in response to the temperature in the boiler as sensed by a sensor bulb 85. The burner cans are mounted in a heat-insulated plenum chamber or housing 78 which is supplied with combustion air at the rearward end from the blower 62, as best shown in FIG. 2. Because the combustion process is continuous and at low pressure, rather than intermittent and at high pressure, as in a piston-type internal combustion engine, a higher combustion efficiency is achieved with less unburned hydrocarbons, oxides of nitrogen, and carbon monoxide being produced.

The hot products of combustion pass from the burner housing 78 into an inlet opening 80 (FIG. 2) adjacent the hot end of a boiler housing or jacket 82 of the boiler 20. The lower end of the jacket 82 is secured around the periphery thereof to a boiler tube supporting control plate 84 of the engine power control system 36 and the control plate is secured to a housing 86 of the positive displacement expander 18 by a plurality of fasteners 88. The control plate 84 serves as a boiler tube support header and is provided with a plurality of openings to receive and support the lower ends of a plurality of relatively small diameter, heat-resistant, high-pressure boiler tubes 90 which extend upwardly in the boiler housing 82 toward the inlet end. The upper ends of the boiler tubes are connected to a common feed-water header 92 adjacent the upper end of the housing and the header is supplied with feed-water from the pump 28 through the feed-water supply line 32. The upper header serves as a feed-water heater (the feed-water being at an inlet temperature of about 281°), and the water leaves the heater and flashes off into superheated steam as it moves down the boiler tubes 90 toward the control plate header 84.

The tubes 90 are arranged in closely spaced, parallel array in a stack or bundle comprising staggered rows of tubes. Because of the requirement for fast start-up of the engine, the boiler tubes have a low mass with a relatively little residual heat capacity and extremely high heat conductance through the tube walls. In a prototype design, ¼-inch stainless steel tubing was used and the tubes were arranged in 19 rows, 17 tubes per row, to provide a total heat transfer area of about 22.9 sq. ft. The boiler tube bundle measures about 9 × 12 inches in transverse cross section, and has an overall height above the control plate 84 mounted on the positive displacement expander of about 18 inches. The tube wall thickness of the stainless steel tubes was 0.050-inch, and a total of 324 tubes was provided. The boiler is capable of a standard design output of 100 horsepower and a maximum acceleration output of 200 horsepower. Nominal design heat load is approximately 1,210,000 BTU's per hour with a boiler output of 1,090,000 BTU's per hour at a steam flow rate of about 840 pounds per hour. The boiler is adapted to operate at a working pressure of 3,500 psia and a temperature of 1,200°F. The total mass or weight of the tubes 90 and the feed-water header 92 is approximately 50 pounds, and the boiler as so designed in combustion with the burner system as described, is able to get up steam very rapidly and cause rotation in the expander within about 45 seconds from a cold start. When operating at the design load, the burner fuel flow is approximately 8.7 gallons per hour (kerosene) with a burner airflow from the blower 62 of 217 SCFM at 20 inches of water pressure.

In order to effectively utilize the heat available from the combustion gases entering the boiler housing 82 through the opening 80, the housing is provided with a plurality of gas flow directing transverse baffles 94 to increase the length of the flow path of the gases, and the combustion process continues to take place in the boiler housing as the gases flow around the baffles back and forth across the tubes **90** in the direction of the arrows shown. Because of the relatively long flow path and greater time for burning, as compared with the short length of time available for combustion in a piston-type internal combustion engine, a higher efficiency burning of the fuel is produced, with a resultant lower pollutant factor in the gas discharged from the exhaust system of the engine. Combustion products are discharged adjacent the upper end of the boiler housing through a discharge outlet **96** into a suitable exhaust system **98.** If required to meet further pollution specifications, further catalytic treatment of the combustion products in the exhaust system may be provided; however, it is believed that the emission of unburned hydrocarbons, oxides of nitrogen, and carbon monoxide in the gases leaving the boiler exhaust outlet **96** is sufficiently low to meet present and future air pollution control specifications.

In accordance with the present invention, the expander **18** is of the positive displacement type and is especially designed to operate at inlet steam temperatures up to approximately 1,200°F. and to pressures up to 3,500 psia. Under these inlet conditions, an expansion pressure ratio of 70 to 1, a volume expansion of about 40 to 1, and a thermodynamic efficiency of 23.4 percent is achieved. The expander **18** is of the sliding vane type and has the advantage of compactness, simplicity, and high efficiency. The control system **36** provides a unique control arrangement which, together with rotor expansion chambers defined between the vanes, allows the steam to flow almost continuously into a clearance volume with little throttling action or pressure loss. The relatively small diameter of the expander housing **86** is selected so that the steam flow velocities are low relative to the Mach number of the steam at the inlet conditions. The peripheral steam velocity at a maximum RPM of the rotor shaft and rotor assembly **15** is only one-tenth of the Mach number for the steam at inlet conditions, and the pressure drop across the control valve section **36** at 3,500 psia inlet pressure is relatively insignificant.

The expander housing is preferably formed of stainless steel and is provided with a cylindrical bore **100** extending between parallel opposite end faces **102,** as best shown in FIGS. **2** and **4.** One end of the bore is closed by a closure member **104** (FIG. **3**) and the opposite end is sealed with a closure member **106** having an elongated axially extending shaft housing **108** projecting from the outer face thereof to enclose the support the rotor shaft **14,** as shown in FIG. **3.** The closure member **104** comprises a circular disc having an inwardly protruding, axially centered, cylindrical bearing portion **110** in coaxial alignment with the rotor shaft **14.** Similarly, the opposite closure member **106** includes an axially centered, inwardly projecting, cylindrical bearing portion **112** which is hollow in the center to accommodate the shaft **14** and is axially aligned therewith. The bore **100** and closure members **104** and **106** define a large, multisection, annular, expansion area which is axially centered with the output shaft **14.** The outer edge portions of the closure members **106** and **104** are secured to the housing **86** by suitable cap screws (not shown) and seal rings of suitable materials are provided to seal between the closure members and the main body of the housing.

Inside the expansion area defined by the cylindrical bore **100,** thee is provided an eccentric, annular liner **114** which is formed of graphite material and has an inner surface coating **115** of vapor deposited, silicon carbide material for withstanding wear and the high temperature of the steam. As shown in FIG. **2,** the inner bore of the liner **114** is oval in transverse cross section and is eccentric in relation to the axis of rotation of the output shaft **14.** The silicon carbide lining material **115** on the inner eccentric surface of the liner is vapor phase deposited and is extremely well suited for the high temperature operation and the sliding friction involved as the vanes of the rotor assembly slide thereon. This material is strong, lightweight, fully dense, extremely hard, has a relatively low coefficient of friction, and is particularly well suited to withstand operating temperatures of 1,200°F. and greater.

In accordance with the invention, a plurality of radially oriented, circumferentially spaced, fluid expansion chambers are formed between the outside surface of a main rotor **116** and the inner bore surface **115** of the eccentric liner **114** between pairs of adjacent vanes carried by the rotor. The main rotor includes an annular outer rim portion and a radial web **118** disposed between the inner end faces of the inwardly projecting bearing portions **110** and **112** of the end closure members. The rotor includes a central hub **120** which is keyed or splined onto the inner end portion of the rotor shaft **14,** as best shown in FIG. **3.** Portions of the inside surfaces of the outer rim **116** of the rotor assembly **16,** extending inwardly from the outer edges of the rim, are provided with vapor phase deposited, silicon carbide coatings designated as **122** (FIGS. **2** and **3**).

As best shown in FIGS. **2** and **4,** the outer rim of the rotor wheel **116** is provided with a plurality of longitudinally extending, circumferentially spaced, radial slots or grooves **124,** each of which receives a set of vanes **126** to define a plurality of expansion chambers around the rotor inside the bore of the liner **114.** In order to provide an effective seal between the adjacent circumferentially spaced, expansion chambers, (as best shown in FIGS. **2** and **4**) each vane set **126** includes an elongated vane **128** having an outer edge **128**a parallel to the rotor axis adapted for sliding contact with the internal surface coating **115** on the bore of the eccentric liner **114.** The vanes **128** are biased radially outwardly in the slots **124** by an undulating vane spring **130** to provide sufficient pressure to establish good sealing contact along the entire outer surface **128*a*** of the vane with the silicon carbide coating **115.**

Each radial slot **124** in the rotor is of a width sufficient to accommodate a pair of relatively shorter end vanes **132** which are biased in opposite, longitudinal directions, as indicated by the arrows in FIG. **4,** towards the end closure members by a biasing spring **134.** Each of the end vanes **132** has an outer surface edge **132*a*** adapted to sealingly engage an annular, end sealing plate **136** preferably formed of vapor deposited silicon carbide material. The vanes themselves (**128** and **132**) are likewise formed of vapor deposited silicon carbide which is well suited to withstand the high temperatures and pressures involved and to resist wear due to the sliding action. From the foregoing, it will be seen that each set of vanes **126** mounted in longitudinal slot **124** of the rotor provides a radial dividing wall between adjacent circumferentially spaced expansion chambers formed between the outer rim portion **116** of the rotor and the coated inner surface **115** of the eccentric liner **114.**

As viewed in FIG. **2,** as the rotor turns in a clockwise direction, the individual expansion chambers under the influence of high pressure, superheated steam, expanded in volume and thereby extract work from the steam to turn the rotor and shaft. The expansion is continuous as the chambers rotate, and the steam pressure decreases at a substantially uniform rate from the 3,500 psia inlet pressure to an exhaust pressure of approximately 50 psia as each expansion chamber reaches a position adjacent the lower end of the expander housing **86.**

In accordance with the present invention, the annular end seal plates **136** are biased inwardly toward the ends of the rotor rim **116** and against the end surfaces **132***a* of the end vanes **132** by an essentially designed pressure balancing seal ring **138.** The balancing seal ring includes an annular outer ring **138***a* and an inner ring **138***b* (FIG. **4**), which rings are interconnected by radial tubelike spokes **138***c*. A plurality of circumferentially spaced balancing chambers are defined by the spokes and a pressure tap **136***a* is provided in the end seal plates **136** adjacent each balancing chamber to pressurize the chamber approximately equal to pressure of the main expansion chambers around the rotor rim **116.** In this manner, the fluid pressure in each balancing chamber in the balancing rings **136** between adjacent spokes **138***c* is approximately equal to the pressure of the main expansion chambers around the rotor rim **116.** In this manner, the fluid pressure in each balancing chamber in the balancing rings **138** between adjacent spokes **138***c* is approximately equal to the pressure in the main expansion chamber around the rotor assembly moving adjacent thereto. The balancing system thus provides for excellent end sealing under all conditions of pressures and temperatures to take care of various expansions and contractions of the components.

In accordance with the present invention, the rotor assembly **16** is supported for rotation on the bearing portions **110** and **112** of the end closures **104** and **106** by a pair of annular, graphite bearings **140** between the bearing surfaces **110** and **112** and the respective coated surfaces **122** on the inside of the rotor rim **116.** The bearings are water lubricated by high pressure water taken dowstream of the main feed-water pump **28** and this high pressure water (3,500 psia) is delivered to the bearing areas through water supply passages **142,** formed in the end closure plates (FIG. **3**), A start-up bearing pump is provided to insure adequate water pressure for the bearings and is available when the engine is being started, and after starting water is bled from the main system for the bearings. The water pressure supplied to the graphite bearings **140** through the supply passages **142** eventually passes into the void space around the inner end of the shaft **14,** the rotor web **118,** and hub **120,** and an outer shaft seal or packing is provided, as at **144,** in the tubular shaft housing **108** of the end closure **106.** The packing seal is cooled with feed-water which passes through an annular passage **146** formed around the seal. After lubrication of the bearings, the water passes from the internal area around the shaft through an exhaust passage **148** and is discharged into a condensate receiver or enclosure generally indicated as **150.**

The receiver is attached to the lower end of the housing **86** by fasteners **151,** as best shown in FIG. **2,** and has an outlet in communication with the exhaust conduit **26** leading to the condenser **24.** The receiver has an upper inlet opening in communication with several of the expansion chambers around the rotor rim **116** at the position of maximum volume of the chambers adjacent the bottom of the housing **86.** The liner **114** is formed with an elongated exhaust opening **114***a* in this region for discharging fully expanded steam, as shown in FIGS. **2,** and the exhaust opening in the liner is in communication with a similar exhaust opening **86***a* formed in the lower wall of the bore **100** in the main housing. As the expansion chambers around the rotor **116** pass over the exhaust openings **114***a* in the liner, the spent steam discharges through the housing exhaust opening **86***a* into the upper inlet of the receiver **150** for return to the condenser **24** through the conduit **26.**

In accordance with the invention, the power control system **36** provides a means for controlling the flow of working fluid admitted to the expander **18** from the boiler **20** without appreciable throttling action or pressure drop in the steam. For this purpose, the baseplate header **84** includes an elongated recess in the underside thereof and indicated as **84***a*, which recess forms a steam chest in communication with the lower ends of the boiler tubes **90** to receive the high pressure steam therefrom. The steam chest is in communication with a plurality of spaced apart steam inlet passages **152** formed in the sloped upper end wall portion of the housing **86,** as best shown in FIG. **4,** and the inlet steam passages direct superheated steam into the upper expansion chambers around the rotor **16** through aligned passages **154** in the liner **114.** The inlet passages are grouped in an area covering a little less than one-half of the total cross section defined by the steam chest recess **84***a* in the control plate header **84** and the passages are arranged in stagger positions in adjacent rows. The total number of steam inlet passages **152** and coaxially aligned passages **154,** that are left uncovered by a movable control plate **156** govern the power output of the engine. By altering the number of passages that are open to admit steam from the boiler **20** into the rotor expansion chambers in main bore **110** of the expander housing **86,** the engine power is regulated, and for this purpose, the control plate **156** is mounted for sliding movement between a fully closed position covering all the passages to a fully open position wherein all the passages are uncovered and open.

The control plate is movable to any number of intermediate positions between a fully open and a fully closed position and because of the staggered arrangement of holes in adjacent rows, the total cross section of the flow area of the uncovered inlet passages is approximately proportional or linear with respect to plate position relative to fully open or closed positions as the case may be. As each chamber around the rotor **116** passes underneath the array of uncovered inlet passages, the chamber is completely filled with steam. The filling process begins as the leading vane **128** passes under the first row of uncovered inlet passages and ends when the trailing vane passes the last row of uncovered passages. Of course the number of rows uncovered is determined by the position of the control plate **156.** When more rows of inlet passages **152** are uncovered by the control plate **156,** for any given rotor speed, the time for filling the rotor expansion chambers increases. This time can be expressed in terms of an angle of rotation through which the rotor travels while each chamber is being filled. By increasing or decreasing this angle (called the admission angle) more or less steam is admitted to each chamber because the pressure of the steam is substantially equal in all of the uncovered inlet passages **152.** The position of the throttle plate **156** in effect controls the flow area through which filling of the rotor expansion chambers takes place and because the area is varied rather than the pressure drop as with a conventional throttling valve, control of the power output of the expander **18** is highly efficient across a wide range of power settings and rotor speeds.

As an example, with the control plate **156** in a fully open position with all of the passages **152** uncovered the admission angle may approximate 15° whereas with the control plate covering all but a fractional portion of the inlet passages in the first row the admission angle may comprise only about 1° of rotor travel. In both cases, however, the pressure drop through the inlet passages between the recess **84***a* in the header and rotor is substantially the same. Moreover the control plate **156** is the only movable part and no other valves are required.

Fluid other than steam can be used to drive the expander **86** which for example, can be used alone at a remote location as an air motor or a hydraulic motor supplied from a separate source of pressurized fluid. Because of the relatively high expansion ratios provided by the expander and the novel control arrangement afforded thereby, many uses as a drive motor are worthwhile. In addition the expander can be used as a compressor or pump by supplying torque to rotate the shaft in an opposite direction. As will be described hereinafter, the novel use of materials in the expander **86** eliminates the need for lubrication other than the working fluid itself and accordingly lubrication problems are simplified resulting in many applications for the expander **86** used as a motor or pump/compressor wherein contamination of the working fluid with lubricants is a problem. One specific application in the use of the expander **86** as a compressor to supply air for paint spraying wherein lubricating oil in the compressed air is not tolerable.

The control plate **156** is provided with a pair of rack-like teeth formations **156***a* (FIG. 3) along opposite side edges, and the toothed rack formations are in meshing contact with driving gears **158** mounted on a rotatable throttle control shaft **160** (FIG. 3). Rotation of a pulley **162** on the outer end of the throttle shaft drives the throttle plate **156** to open or close the inlet ends of the passages **152.** Rotation of the throttle control shaft **160** in opposite directions is effective to increase or decrease the steam flow and, accordingly, the power output of the heat engine **10** by increasing or decreasing the number of coaxial flow passages **152** and **154** available for directing steam from the boiler to the positive displacement expander **86.**

After expansion of steam in the expansion chambers around the rotor, the spent steam from the receiver **150** is directed through the conduit **26** to a lower header **164** of a cooling condenser **24** somewhat similar to an automobile radiator. The condenser **24** includes a plurality of cooling tubes **166** which may be of the inverted U-shaped type, as shown. The lower ends of the U-tubes are connected to the inlet header **164,** and the upper ends are connected to a second or discharge header **168.** A plurality of cooling fins **170** are provided to facilitate the dissipation of heat in the condenser.

Because the working fluid of the heat engine **10** is water which expands upon freezing in commonly encountered ambient temperatures, the freeze protection system **34** is designed to automatically protect the engine components against damage from such freezing. The freeze protection system **34** uses ammonia gas as an antifreeze agent, and the gas is supplied from an ammonia storage tank **172** and injected into the lower header **168** of the condenser **24** by means of a temperature sensitive control valve **174** through an ammonia supply conduit **176** having a check valve **178** therein. The temperature control valve **174** is sensitive to ambient temperature conditions, and as the temperature drops below the freezing level, when the engine is not operating, the valve opens to admit ammonia from the tank **172** into the liquid water contained in the condenser **24,** conduit **26,** and the lower portion of the receiver **150.** The ammonia gas is compatible with working fluid temperatures of 1,200°F., whereas other common antifreeze agents, such as alcohols and glycols, are not usable at these high temperatures because of chemical cracking and resultant carbonaceous deposits in the boiler and other components. On the other hand, chemical cracking of the ammonia gas dissolved in the working fluid does not produce any such carbonaceous deposits and results only in the formation of hydrogen and nitrogen gas. The freeze protection system includes means for synthesizing these gases back into ammonia to replace the ammonia lost because of the chemical cracking which occurs during operation of the engine, mainly because of the inability to separate out all of the ammonia from the water during start-up and initial heating up of the engine. The temperature control valve **174** is set up so that when the ambient temperature goes below 32° F. and the engine is not operating, the valve opens to admit ammonia gas into the working fluid in the condenser in an amount sufficient to protect against freezing down to a temperature of approximately −40°F. This requires a mixture comprising approximately 22 percent ammonia by weight in the working fluid; however, the ammonia gas is dissolved readily in the water so that the entire volume of water in the system rapidly reaches an equilibrium concentration of aqueous ammonia.

During start-up and initial heat-up of the engine after a dosage of ammonia has been ejected into the working fluid, the aqueous ammonia vaporizes and eventually some of the gas is chemically dissociated into nitrogen and hydrogen gas because of the high temperatures and pressures. These gases do not readily recombine to form ammonia but instead are collected in an upper header **178** which is connected to the header **168** through a plurality of tubes **180.** The nitrogen and hydrogen gases collecting in the upper header **178** are drawn off through a line **182** and are pumped via a nitrogen-hydrogen gas pump **184** into a gas storage tank **186.** A suitable check valve **188** is provided to prevent reverse flow.

A pressure-sensitive switch **190** is associated with storage tank **186,** and when the pressure reaches a level of about 3,000 psi the pressure switch energizes a solenoid operated check valve **192** provided in the tank exit line **194** and also energizes a motor **195** driving a synthesis recirculating pump **196.** The line **194** supplies dissociated nitrogen and hydrogen gas to a catalytic reactor **200** wherein the gases recombine to synthesize ammonia. The gas mixture leaving the reactor via a conduit 202 contains about 40 percent ammonia gas and this gas is condensed by cooling in a cooler 204 to the liquid condensation temperature of ammonia. Condensed out liquid ammonia is separated from the non-condensed gas and is collected in an ammonia trap 206. The remaining, noncondensed gases are recirculated and fed to the inlet side of the recirculating pump 196. The pump recirculates the gases to mix with incoming gases in the line 194 and the mixture of old and new gases passes through the catalytic converter 200 via a line 210. By recirculating the noncondensed gases through the catalytic reactor 200, further synthesis of ammonia is obtained and the ammonia so formed collects in the trap and is then pumped into the storage tank 172 by the pump through the line 214. The synthesized ammonia is stored in the tank until it is again needed for injection as antifreeze into the engine system.

Once the ammonia synthesis process is started in the reactor, the process becomes exothermic, so that little or no additional heat is required for the catalytic bed of the reactor. Moreover, the catalysts used in the reactor are of the type which are readily replaced, should the catalyst become contaminated by excessive water vapor. However, excess water vapor is not usually a problem as the level is normally kept to less than 2/10 of 1 percent by volume in the gases circulated in the ammonia synthesis system. During initial start-up of the engine, the catalytic reactor 200 is heated up to about 1,000°F. by means of a heating system, generally indicated at 216, which takes heated exhaust gases from the boiler exhaust 98 and from a duct 218. The heated gases are initially directed through a branch conduit 220 into heat exchange contact with the catalytic reactor 200. A temperature sensitive control damper 222 is provided to initially direct the hot gases from the supply duct 218 to the reactor and thereafter through a bypass duct 224 when the reactor temperature reaches a level to provide self-sustaining exothermic reaction.

In general, the ammonia synthesis system is designed to take care of an operation environment wherein an extremely ambient temperature of about −40° is encountered. The synthesis system is capable of producing enough ammonia from the cracked nitrogen and hydrogen gas to continuously provide an ample supply of gas for freezing protection of the water in the system down to ambient temperatures of −40°F. Also, the ammonia synthesis system is designed to provide a syntesis rate high enough so that the rate of ammonia protection can satisfy a severe cooling condition should the engine be shut down after a short duration run in an ambient temperature down to −40°F. The synthesis system utilizes very little power and the use of ammonia as the antifreeze rather than glycols or alcohols does not pose carbonaceous deposit problems.

In accordance with the invention, the heat engine 10 is initially started up in an automatically controlled starting sequence similar to that employed in modern jet aircraft engines wherein the vehicle operator merely turns on a starting switch 225 and waits for an automatically controlled starting sequence to take place. When the starting sequence is initially activated, the accessory drive motor 64 is powered from the battery and this rotates the fuel pump 70, the combustion blower 62, the bearing lubrication pump 68, and the auxiliary feed-water pump 72. The auxiliary feed-water pump 72 fills the boiler tubes 90 through a bypass line 91 and check valve 93 and begins to pressurize the boiler to a lower than normal operating pressure. This initial or start-up pressure is below the normal operating pressure of 3,500 psi and is determined by the frictional forces required to start rotation of the rotor assembly 16 of the expander 18. As soon as the water pressure is raised to a selected pressure level as sensed by a bulb 87 in the header 92, a starting fuel valve 85 (FIG. 6) is closed and fuel is supplied to the burners 76 through the manifold 74. The electrical ignition spark system 75 is activated for a timed interval and after light off of the burners, the hot products of combustion flow through the boiler jacket 82 and over the partially filled boiler water tubes 10, as previously described. As steam is formed from the water and the temperature rises, the stream is passed through the partially opened control plate 156 to initiate rotation of the rotor 16 in the idle mode. When the rotor assembly 16 begins to rotate, the main feed-water pump and condenser fans begin to turn, and the engine system begins to build up to full normal operating pressure, which buildup is controlled by a thermostatic bypass valve 95 in the starting feed-water pump conduit 91. The valve is controlled by a sensor 97 in the boiler and eventually the bypass line is closed when full pressure is attained. During the starting sequence, fuel and airflow are controlled in the burner assembly by the boiler thermal sensing device 85 so that the boiler does not become overheated. After starting is completed the device 85 helps maintain an even temperature in the boiler by modulating the fuel supply through the fuel control 71.

During cold weather starts, the procedure is similar except that ammonia antifreeze is vaporized and cracked into nitrogen and hydrogen gas. The ammonia synthesis system 34 is then brought into operation to return the chemically cracked gas back to ammonia for replacement in the ammonia storage tank 172. After shutdown of the engine, a fresh charge of ammonia is injected into the condenser system, as described, to prevent freezing.

During the start up, the lubricating pump 68 is running and delivers water through a conduit 101 and check valve 103 to a bearing supply line 105 connected to the bearing passages 142 in the expander 86 through fittings 143 (FIG. 3). This insures lubrication of the rotor system during the start before full fluid pressure is attained. After rotation begins the main feed-water pump 28 is operative and supplies the lubrication system through a conduit 107.

In operation, opening of the control plate 156 to uncover more steam admission passages 152 and 154 produces the effect of an increase in the size of the clearance volume. This results in an increase in the average pressure acting on the vane sets 126 in the expanding chambers of the rotor 116. On opening of the control plate, additional steam is supplied from the small inventory of the steam in the boiler tubes and steam chest 84*a*. This loss of steam is almost instantaneously made up by the automatic regulation of the bypass valve 95 which then supplies additional feed-water. Feed-water flows directly through the feed-water pump at a higher rate and this, in turn, cools the inlet end of the boiler and the thermal sensing device 85 increases the amount of fuel flowing to the burners. Both the boiler and expander are capable of handling temporary overloads needed for rapid acceleration under normal driving conditions.

Because of the low pressure, relatively slow combustion in the burner system and boiler, the production and emission from the engine of carbon monoxide, oxides of nitrogen, and unburned hydrocarbons is extremely low, in comparison with internal combustion engines presently on the market. Moreover, because of the relatively long combustion interval, as compared to a high speed piston-type internal combustion engine, the fuel combustion is more complete in terms of complete oxidation of the hydrocarbons.

Although the present invention has been described by reference to only a single embodiment thereof, it will be apparent that numerous modifications and embodiments may be devised by those skilled in the art, and it is intended by the appended claims to cover all modifications and embodiments which fall within the true spirit and scope of the present invention.

What is claimed as new and desired to be secured by Letters Patent of the United States is:

1. A heat engine comprising boiler means for increasing the heat energy of a working fluid; positive displacement expander means for using said heat energy of said fluid for useful work, said expander comprising a housing having a cylindrical bore and a pair of parallel opposite end walls, a rotor having ends facing said walls and mounted for rotation in said bore, a liner of wear resistant material mounted in said bore and having a varying wall thickness defining an eccentric fluid chamber around said rotor, said rotor including a wheel with a plurality of radial slots therein, a radial vane assembly in each of said slots projecting outwardly thereof and defining said eccentric chamber into a plurality of separate variable volume fluid expansion chambers around the periphery of said wheel within said liner, said vane assembly including a first vane biased radially outwardly in said slot toward an inside wear surface of said liner and a pair of end vanes biased outwardly of said slot toward said end walls of said housing, a pair of annular seals between said end walls of said housing and adjacent ends of said rotor, at least one of said seals comprising an annular pressure ring bearing against an adjacent end of said rotor and a seal balancing ring between said pressure ring and an end wall of said housing, said seal balancing ring comprising a pair of inner and outer radially spaced concentric seal rings joined together by a plurality of spaced apart radial divider spokes defining a plurality of pressure balancing chambers, a plurality of ports in said pressure ring for supplying pressurized fluid from expansion chambers around said rotor wheel to adjacent pressure balancing chambers behind said pressure ring, and fluid control means between said boiler means and said eccentric chamber in said housing for regulating the filling of said expansion chambers with said high energy fluid.

2. The heat engine of claim **1** wherein said fluid control means comprises a plurality of fluid passages extending between an outlet end of said boiler means and at least one expansion chamber around said rotor, and means for opening and closing said passages to increase or decrease the number of passages filling a particular expansion chamber as it moves into and out of communication with said passages.

3. The heat engine of claim **2** wherein said passages are arranged in staggered relation in parallel rows and communicate with said eccentric chamber through open ends disposed in a substantial arcuate shaped filling area on said inside wear surface of said liner, said control means including means movable to open and close opposite ends of said passages to increase and decrease the effective cross-sectional area of fluid flow between said boiler means and said eccentric chamber.

4. The heat engine of claim **1** wherein said inside wear surface of said liner is coated with silicon carbide material.

5. The heat engine of claim **1** wherein said vanes have outer edges for sliding contact with said wear surface of said liner comprising silicon carbide material.

6. The heat engine of claim **1** wherein said housing includes at least one annular bearing surface for said rotor, and lubrication means for supplying said working fluid in liquid form for lubrication of said bearing surface.

7. The heat engine of claim **6** wherein said lubrication means includes pump means receiving fluid upstream of said boiler means for supplying liquid to said bearing surface.

8. The heat engine of claim **6** including bearing means comprising one bearing surface of silicon carbide and a cooperating bearing surface of graphite for supporting said rotor for rotation in said housing, said lubrication means supplying said working fluid between said surfaces through said graphite.

9. The heat engine of claim **6** wherein said pump means comprises a feedwater pump for supplying working fluid to said boiler means and a bearing pump driven independently of said feedwater pump.

10. A heat engine utilizing steam as a working fluid comprising feedwater pump means for increasing the pressure of said fluid; boiler means for supplying heat energy to said fluid received from said pump means; positive displacement, fluid expander means for receiving fluid from said boiler means and producing useful work therefrom, said expander means including a rotor having a plurality of spaced apart radially extending vanes defining a plurality of separate variable volume expansion chambers around said rotor, a housing having a cylindrical bore coaxially aligned and encircling said rotor, a liner in said bore having a hardened inside wear surface for contact with outer edges of said vanes and having a variable wall thickness for changing the volume of each chamber as it moves around said bore; condenser means for receiving expanded fluid from said expander means and condensing the same to liquid for supplying to said feedwater pump means, starting feedwater pump means driven separately of said expander, and bypass conduit means communicating between said condenser means and said boiler means through said starting feedwater pump means for delivering fluid under pressure from said starting feedwater pump to said boiler means during initial startup of said heat engine, and control means for controlling the volume flow of fluid from said boiler means to said expander means.

11. The heat engine of claim **10** wherein said control means includes passage means for filling said expansion chambers with working fluid in an area defined around a portion of said inside wear surface of said liner swept by said vanes.

12. The heat engine of claim **11** wherein said last mentioned means comprises a plurality of spaced apart filling passages having outlets located in said area of said inside wear surface and means for opening and closing said passages to expand and contract the ratio of the area swept by each vane during filling through open passages and the total area swept by each vane during each revolution of said rotor.

13. The heat engine of claim **10** including fuel burner means for supplying heated gases for passage through said boiler means to heat said working fluid, said control means including fuel control means for controlling the rate of fuel supplied to said burner means in relation to the volume of working fluid flowing into said expander means.

14. The heat engine of claim **10** wherein said fuel control means includes means sensitive to the temperature of said boiler means for controlling fuel flow to said burner means.

15. The heat engine of claim **10** including means for injecting anti-freeze fluid into said working fluid in response to ambient temperature conditions.

16. The heat engine of claim **15** wherein said anti-freeze comprises ammonia and including synthesis means interconnected between said condenser means and said anti-freeze injection means for removing gaseous fluids from the liquid in said condenser means and synthesizing ammonia therefrom for use in said anti-freeze injection means.

17. The heat engine of claim **16** wherein said synthesis means includes catalytic reactor means and means for supplying heated exhaust gases from said boiler means to said reactor means.

* * * * *