United States Patent

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[54] **COMFORT CONDITIONING SYSTEM**
24 Claims, 9 Drawing Figs.

[52] **U.S. Cl.** **165/2,** 165/26; 62/183
[51] **Int. Cl.** **F25b 13/00**
[50] **Field of Search** .... 165/22, 23, 26.2, 42—44; 62/183

[56] **References Cited**
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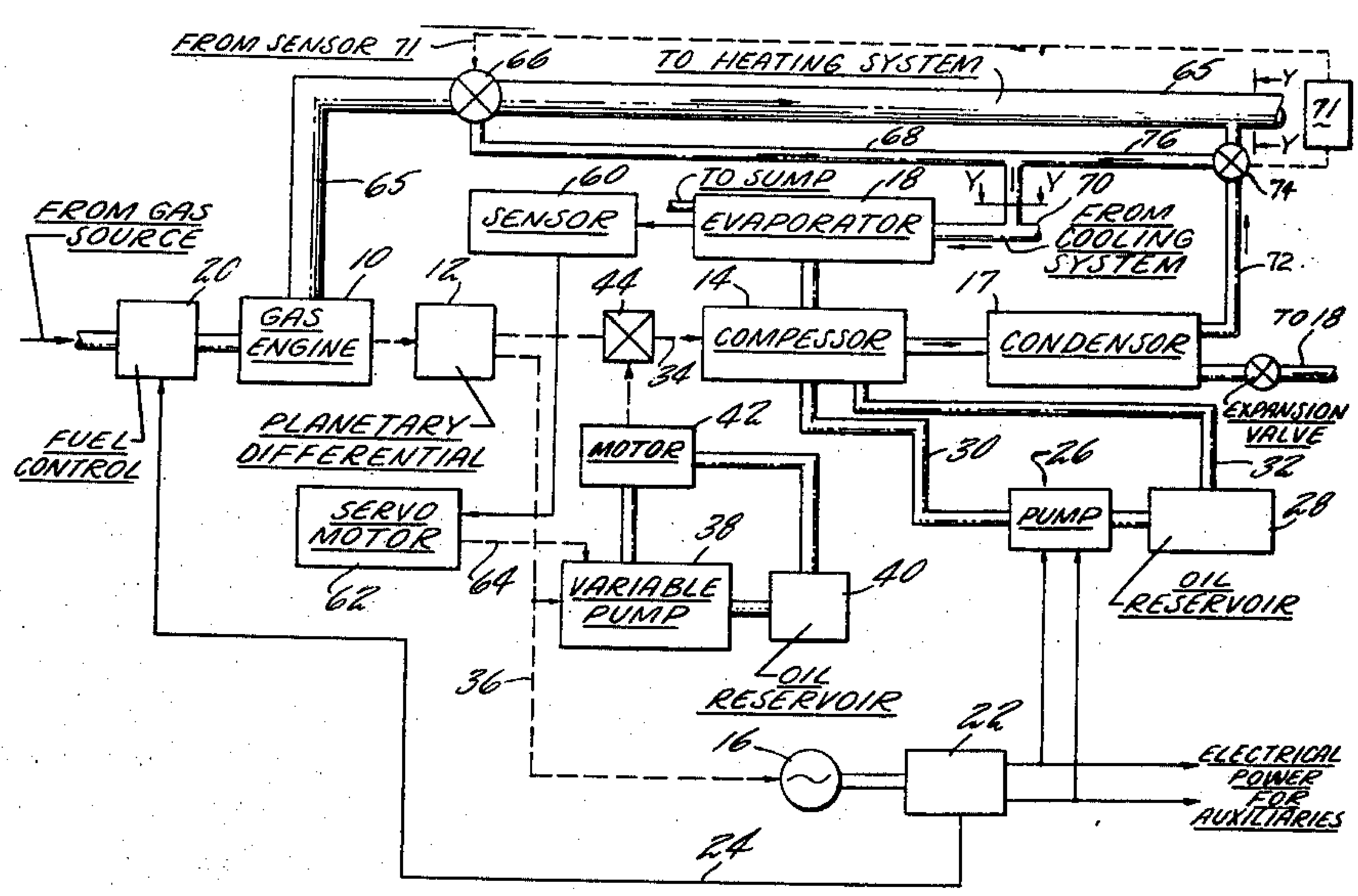
ABSTRACT: The heating and cooling of building structures is accomplished by driving an air conditioning system compressor with an internal combustion engine. In the heating mode, engine waste heat and condensor rejected thermal energy are used to supply heat to the structure and the total thermal energy rejected by the condensor is increased by feedback of energy from the condensor to the evaporator to create an artificial cooling load. Heating is, accordingly, accomplished without reversal of the refrigerant system. In addition, the present invention is self-sustaining as long as fuel is supplied to the engine by virtue of inclusion of a generator which is driven by the engine; means for adjusting the engine shaft power split between the generator and compressor in response to evaporator conditions being included.



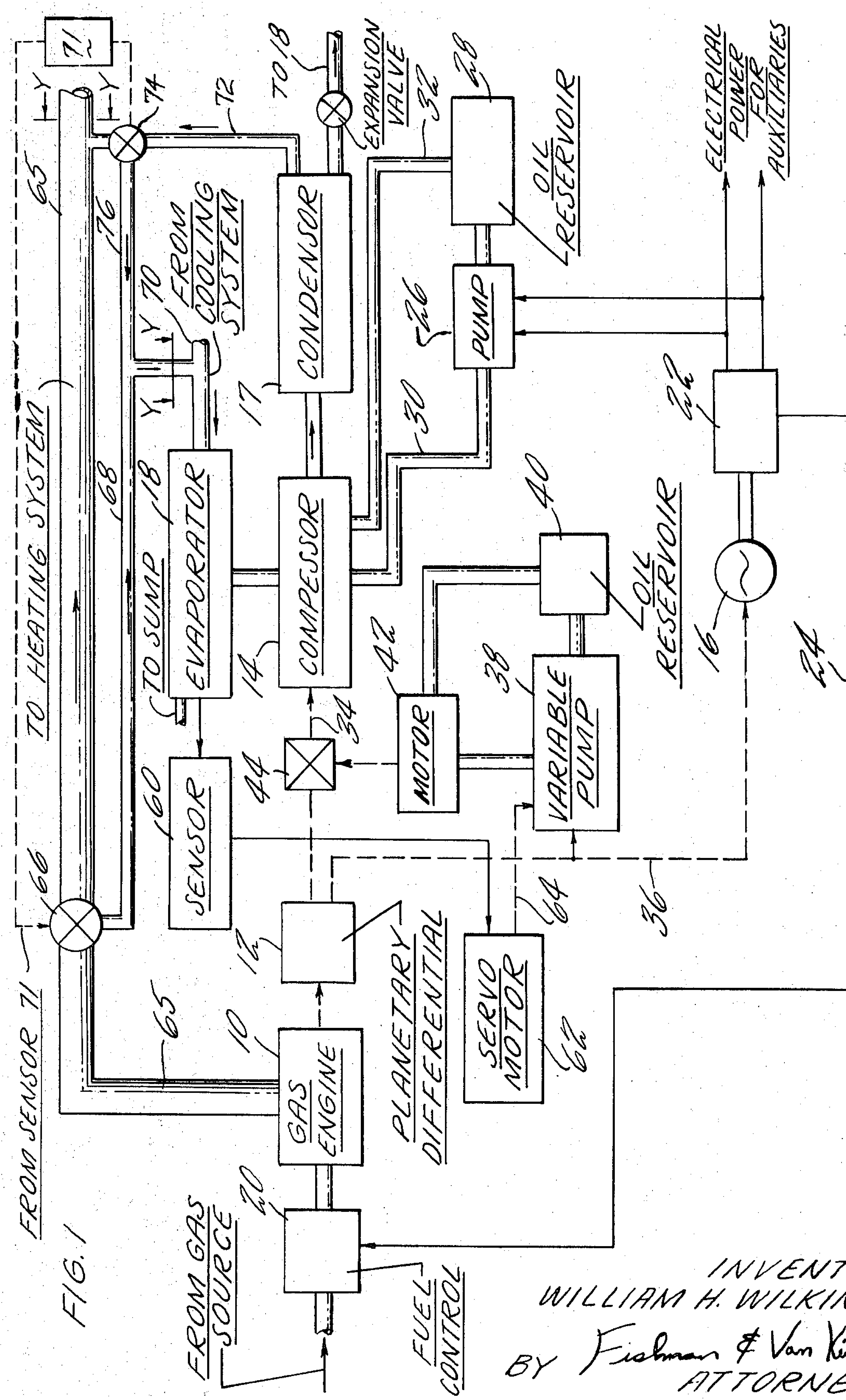


FIG. 1

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FIG. 2

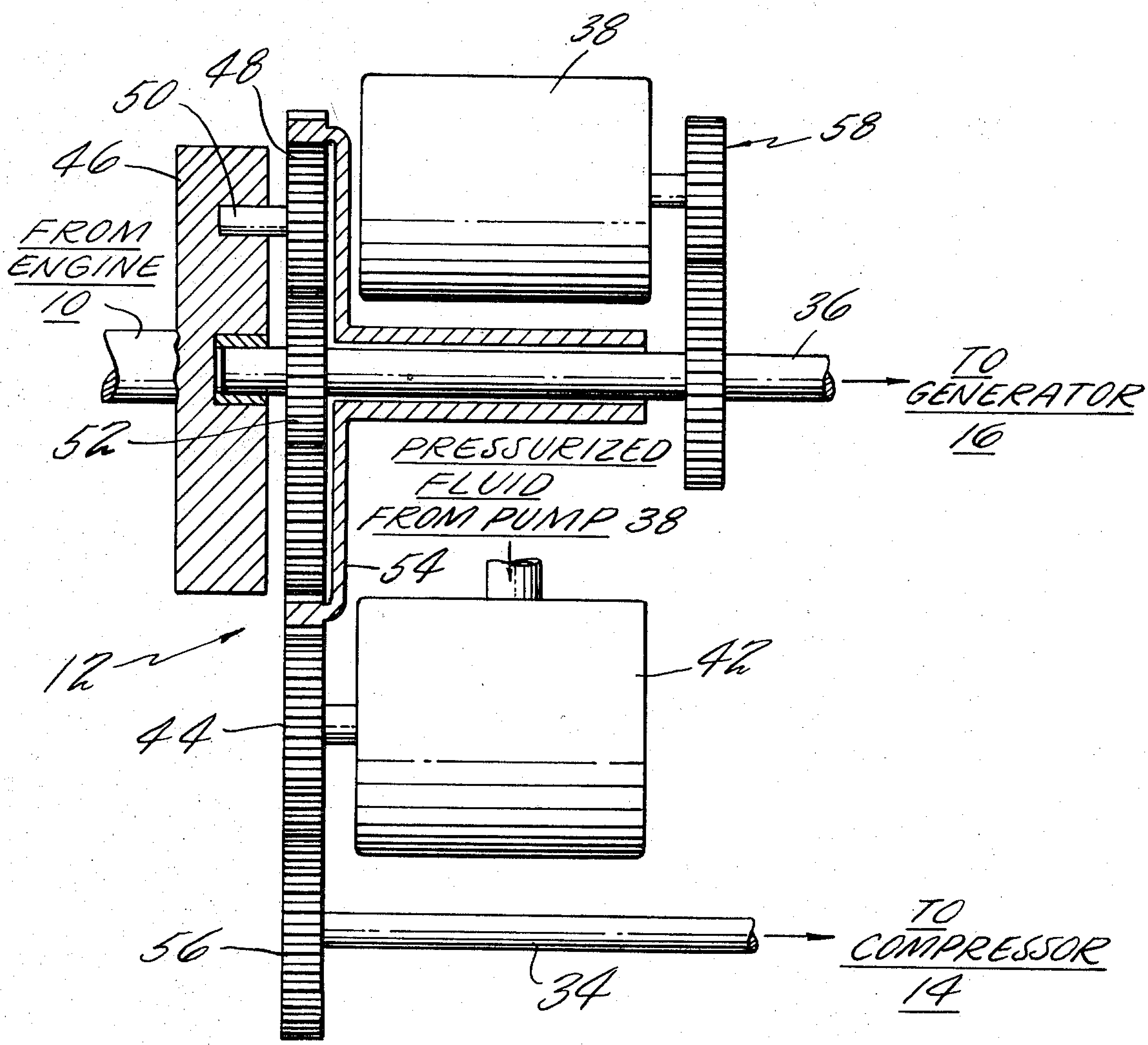

38
48
50
46
58
FROM ENGINE 10
36
TO GENERATOR 16
52
PRESSURIZED FLUID FROM PUMP 38
54
12
42
44
56
TO COMPRESSOR 14
34

FIG. 5

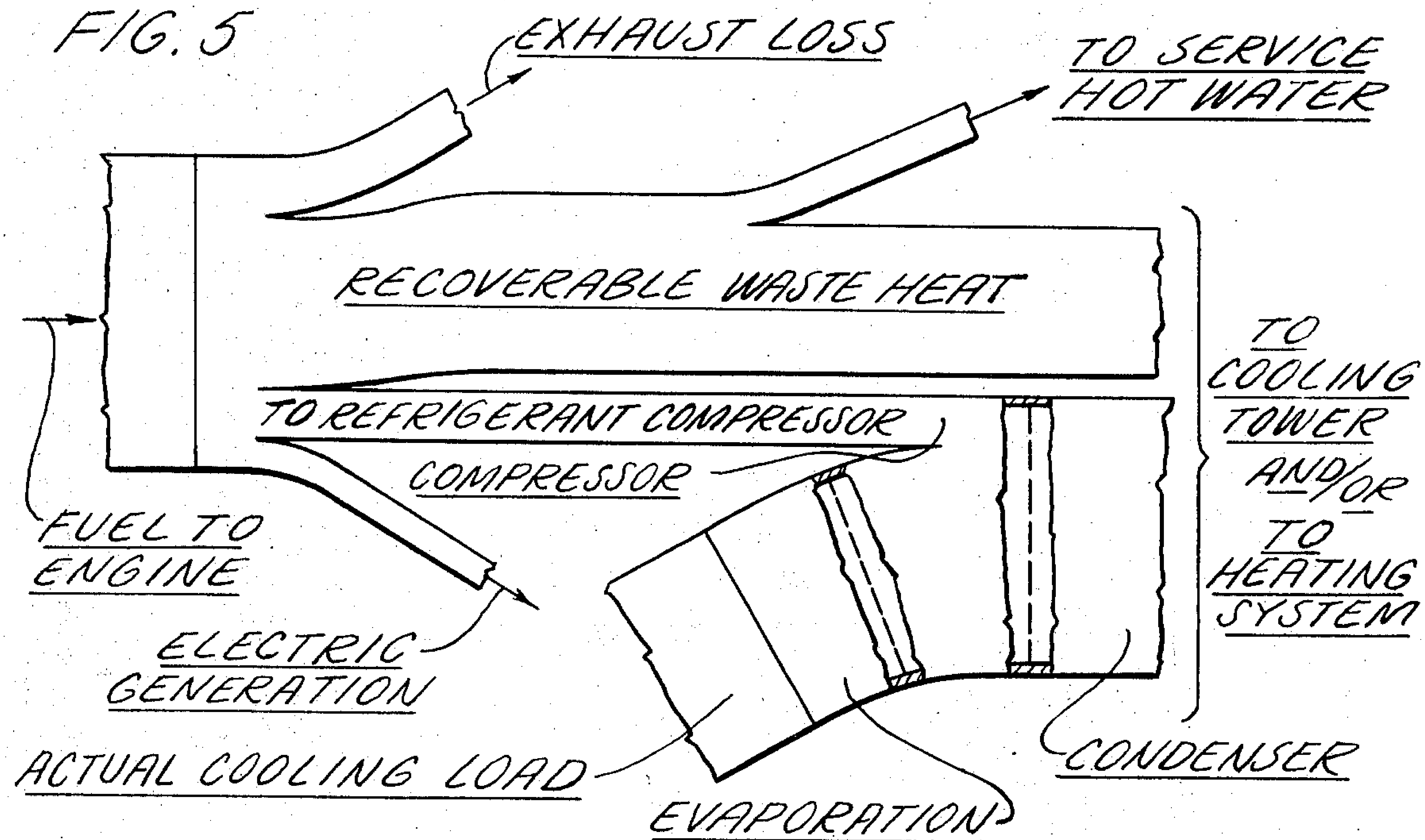

EXHAUST LOSS
TO SERVICE HOT WATER
RECOVERABLE WASTE HEAT
TO COOLING TOWER AND/OR TO HEATING SYSTEM
TO REFRIGERANT COMPRESSOR
COMPRESSOR
FUEL TO ENGINE
ELECTRIC GENERATION
ACTUAL COOLING LOAD
EVAPORATION
CONDENSER

FIG. 6

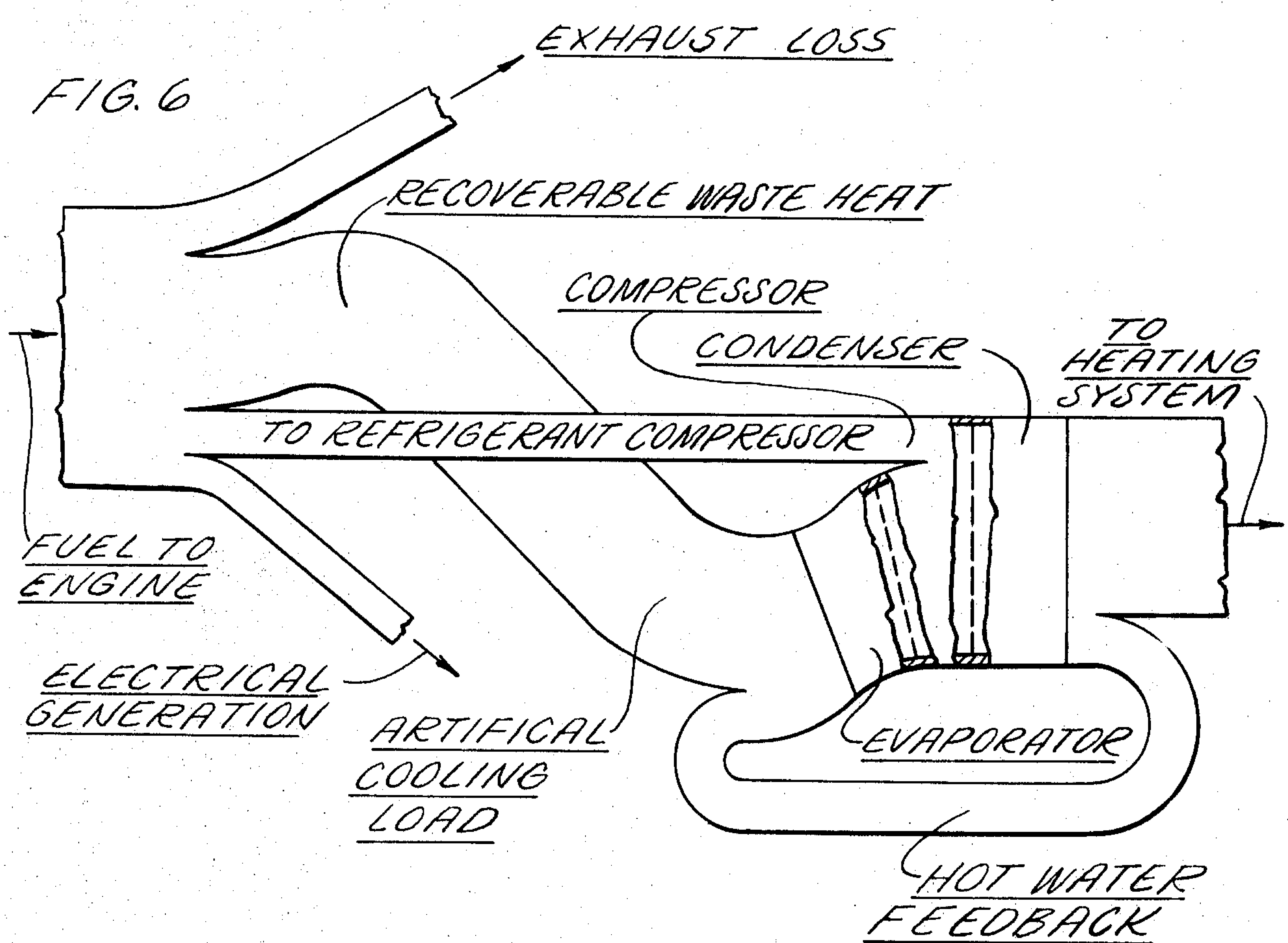

EXHAUST LOSS
RECOVERABLE WASTE HEAT
COMPRESSOR
CONDENSER
TO HEATING SYSTEM
TO REFRIGERANT COMPRESSOR
FUEL TO ENGINE
ELECTRICAL GENERATION
ARTIFICAL COOLING LOAD
EVAPORATOR
HOT WATER FEEDBACK

FIG. 3

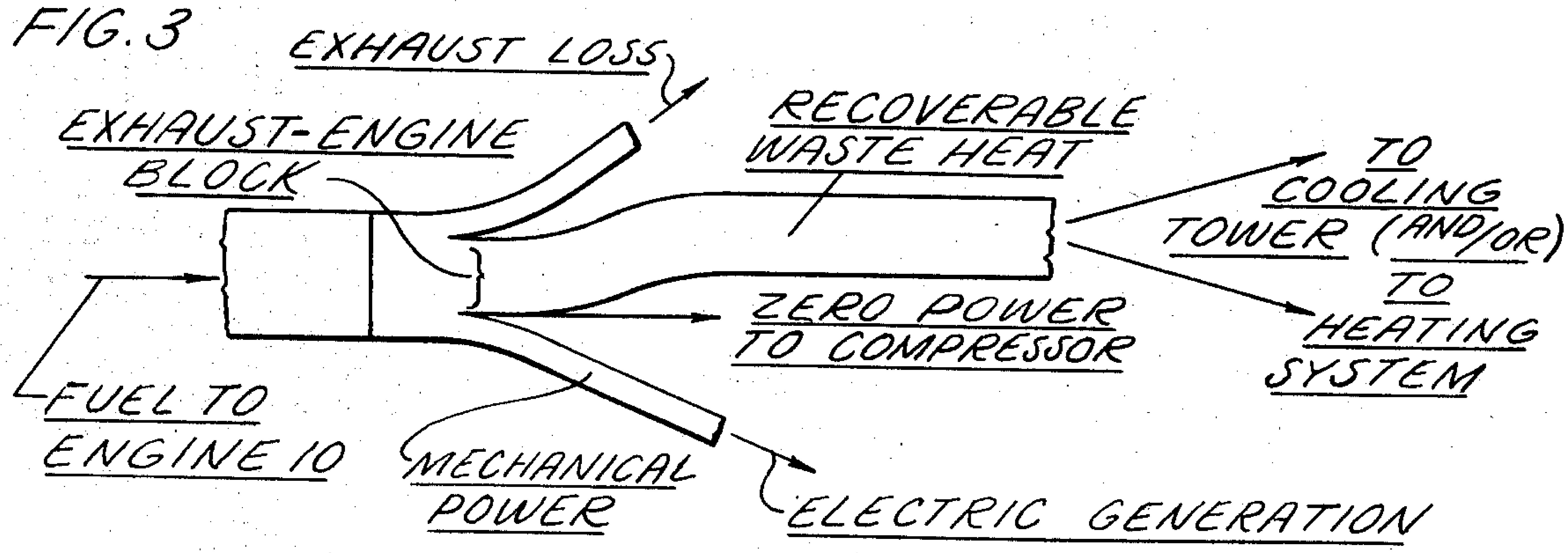


FIG. 4

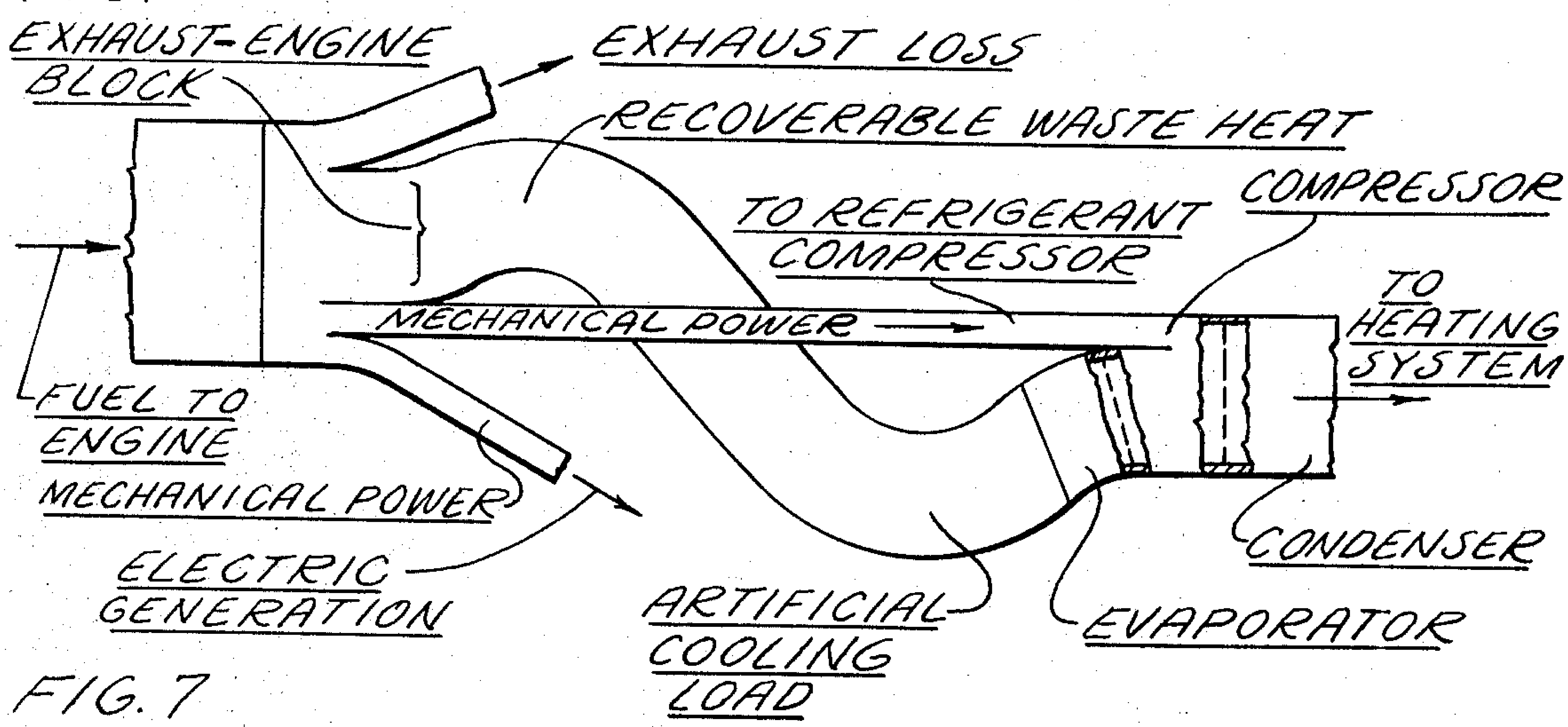


FIG. 7

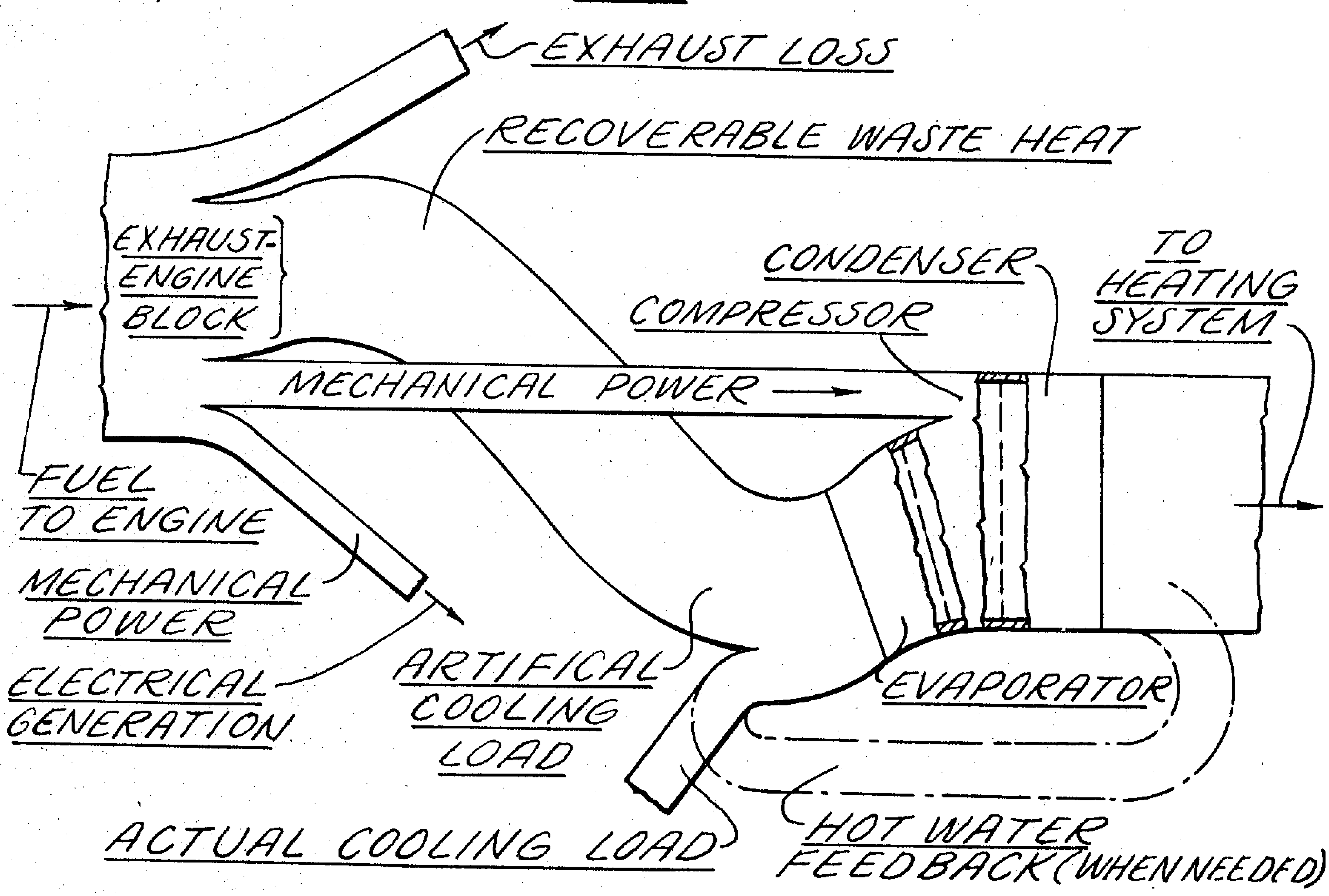
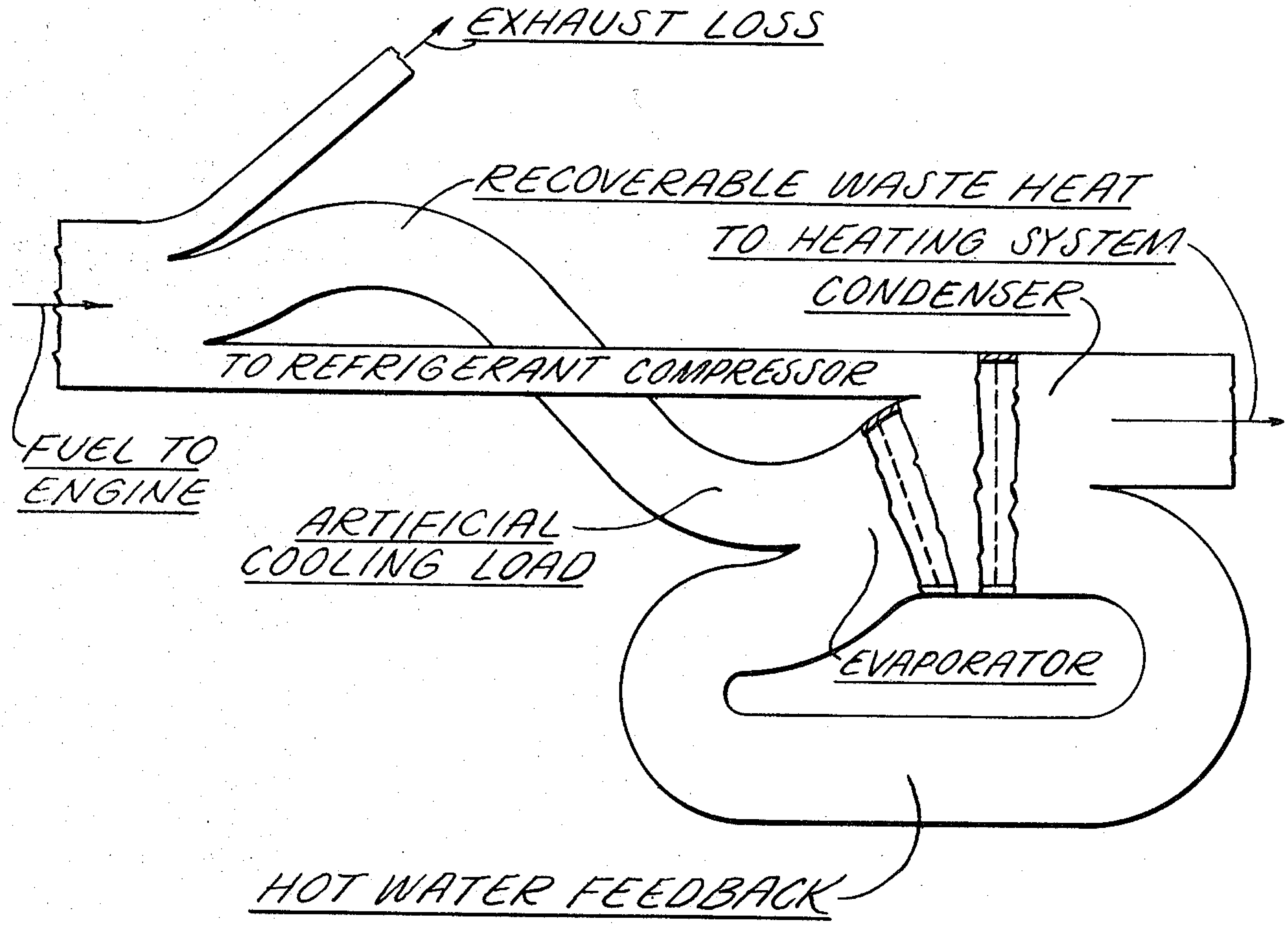


FIG. 8


EXHAUST LOSS
RECOVERABLE WASTE HEAT
TO HEATING SYSTEM
CONDENSER
TO REFRIGERANT COMPRESSOR
FUEL TO ENGINE
ARTIFICIAL COOLING LOAD
EVAPORATOR
HOT WATER FEEDBACK

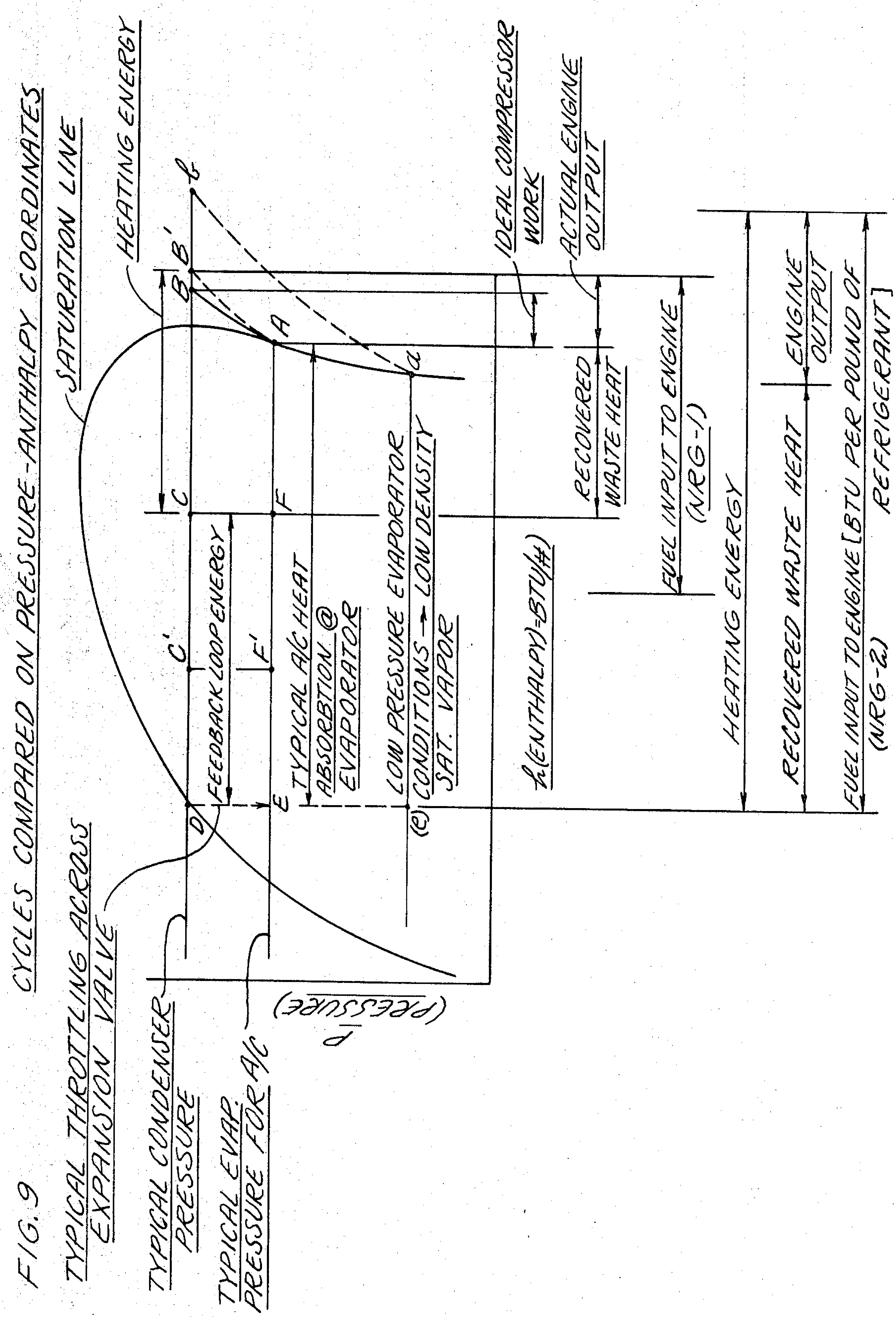

CYCLES COMPARED ON PRESSURE-ANTHALPY COORDINATES
SATURATION LINE
HEATING ENERGY
IDEAL COMPRESSOR WORK
ACTUAL ENGINE OUTPUT
TYPICAL THROTTLING ACROSS EXPANSION VALVE
TYPICAL CONDENSER PRESSURE
TYPICAL EVAP. PRESSURE FOR A/C
FEEDBACK LOOP ENERGY
TYPICAL A/C HEAT ABSORBTION @ EVAPORATOR
LOW PRESSURE EVAPORATOR CONDITIONS → LOW DENSITY SAT. VAPOR
h (ENTHALPY)=BTU/#
P (PRESSURE)
RECOVERED WASTE HEAT
FUEL INPUT TO ENGINE (NRG-1)
HEATING ENERGY
RECOVERED WASTE HEAT
ENGINE OUTPUT
FUEL INPUT TO ENGINE [BTU PER POUND OF REFRIGERANT] (NRG-2)
A
B
B'
B''
C
C'
D
E
(e)
F
F'
a
b

FIG. 9

COMFORT CONDITIONING SYSTEM

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to the comfort conditioning of building structures. More particularly, the present invention is directed to an independent regenerative internal recovery system for use in the heating and cooling of structures. Accordingly, the general objects of the present invention are to provide novel and improved methods and apparatus of such character.

2. Description of the Prior Art

Presently the heating and cooling of buildings is typically achieved by use of electric air conditioning equipment coupled with combustible fuel fired heating equipment. In addition, heating and cooling systems have been proposed, and in some cases implemented, wherein the energy extracted from the same fuel has been employed for both heating and cooling. These prior art systems have, as a general rule, utilized a driving engine. In their most basic form, such systems employ a gas engine to drive an air conditioner and, in addition, employ a gas-fired heater to supply thermal energy. In addition, so-called total energy systems have been proposed wherein the combustion of a fuel has been at least partly utilized in the generation of electricity in order to provide the security against power failure that is implicit in the isolation of an entire structure from the electric utility. In the total energy systems, a gas engine may be used to drive a generator to produce electricity and an attempt is made to recover waste heat for heating and cooling. Further engine drive systems are variations on the known reverse cycle heat pump, the waste heat emanating from the engine as it drives a refrigeration compressor being recovered and used for heating.

There is, of course, a long standing desire to improve the efficiency of and either lower or justify the initial cost of the more sophisticated of the above briefly described systems. In addition, "limited energy systems," wherein the building structure would not be isolated from the electric utility but wherein the heating-cooling system would be self-sustaining as long as a combustible fuel was supplied thereto have also been considered desirable. With particular respect to prior art heat pump systems, experience with such systems wherein a gas engine is the driving source have demonstrated disproportionately large electrical energy operating costs to the extent that the power required from an electric utility for fan and pump drives represents a power cost which is in excess of the cost of the combustible fuel delivered to the driving engine. Obviously, therefore, the ability to drive as many of these auxiliary fan and pump devices as possible directly or indirectly from the comparatively low operating cost gas engine would provide a substantial economic advantage if this could be accomplished efficiently.

The above-mentioned typical prior art electric air-conditioning-gas-fired boiler comfort control systems have posed a substantial problem for utility companies. That is, the high demand for gas during the heating season and the large demand for electricity during the cooling season has the adverse effect of unbalancing yearly distribution loads and in cases of unusual weather conditions, imposes excessive demand on distribution and/or storage capabilities. Accordingly, a desire to employ a gas engine to economically accomplish year round comfort conditioning is present; a system employing a gas engine improving the balance between summer and winter gas loads. It is to be noted that engine driven heat pumps satisfy the criteria of year-round use but, due to the complexity of equipment associated with reversal of a refrigerant cycle, are burdened by a high first cost and a comparatively low efficiency.

SUMMARY OF THE INVENTION

The present invention overcomes the above-discussed and other disadvantages and limitations of the prior art and, in so doing, provides novel and highly efficient comfort conditioning systems for the heating and cooling of structures. The systems of the present invention accomplish these objectives without employing expensive, reverse cycle, classical heat pumps. In accordance with the present invention, waste heat is recovered from a gas engine driving an air conditioning compressor. This waste heat or a portion thereof, with the addition of a portion of the heat rejected from the condenser of the air conditioning system, is supplied as an artificial cooling load to the air conditioning system evaporator in a heating operative mode. The total thermal energy rejected by the condenser is thus increased and this energy is utilized for heating the structure. In accordance with the present invention, heating is accomplished without reversal of the refrigerant system interconnecting the evaporator and condenser.

Also in accordance with the present invention, a self-sustaining comfort conditioning system may be provided wherein an electrical generator is driven by the gas engine. The foregoing is achieved through the use of a differential connection between a gas engine prime mover and both an electrical generator and a refrigeration compressor. A novel control system allows the generator to supply electrical energy at constant frequency for the auxiliary pumps and fans associated with the heating system and, at the same time, provides for speed modulation of the compressor to achieve capacity control. In accordance with the present invention, since compressor speed can be very small, an increased modulation range is obtained without unloading valves, the system inherently unloads itself for starting and the compressor shaft can be stalled for idle operation. The incorporation of the electrical generator enhances the heating capability of the system since the increased power generated by the engine to drive the generator releases proportionately increased waste heat from the engine which can be recovered for heating; such waste heat being utilized either directly or as an artificial cooling load for the air conditioning system evaporator.

BRIEF DESCRIPTION OF THE DRAWING

The present invention may be better understood and its numerous advantages will become apparent to those skilled in the art by reference to the accompanying drawing wherein like reference numerals refer to like elements in the various FIGS. and in which:

FIG. 1 is a block diagram of a preferred embodiment of a comfort conditioning system in accordance with the present invention.

FIG. 2 is a side elevation view of a differential connection between the gas engine and other rotating machinery which may be used in the embodiment of FIG. 1.

FIGS. 3—8 are schematic representations depicting various modes of operation of the comfort conditioning system of the present invention.

FIG. 9 is a pressure enthalpy diagram which illustrates the operation of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

With reference now to FIG. 1, a preferred embodiment of the present invention comprises a gas engine **10** which drives, preferably through a planetary differential gear arrangement **12**, a compressor **14** of a refrigeration system and an electrical generator **16**. The refrigeration system also comprises in part a condenser **17** and an evaporator **18**. It is contemplated that the system of FIG. **1** will, via the output power from generator **16**, provide sufficient electrical power to drive all of the auxiliary pumps, fans, and other devices of the comfort conditioning system as long as fuel is supplied to gas engine **10**. The load imposed by these auxiliary pumps, fans and other devices is essentially a constant power, constant speed generator load and, therefore, imposes a substantially constant torque demand upon engine **10**. Although compressor **14** will operate over a range of speeds for modulation during either heating or cooling, the compressor torque is nearly constant. Thus, with some relatively minor adjustments required between the heating mode and the air conditioning mode, the nearly constant torque requirements of the system dictate a differential connection, as furnished by planetary differential gear 12, between gas engine 10 and the other system components driven thereby.

While not limited thereto, it is contemplated that gas engine 10 will be a conventional reciprocating piston, internal combustion engine which has been modified to run on natural gas fuel. The fuel for engine 10 will be supplied from a gas source, not shown, via a conventional fuel control device 20 which includes a solenoid operated fuel supply valve. The system of the present invention is self-regulating and thus the rate at which fuel is delivered by fuel control 20 to engine 10 will be automatically controlled in the manner to be described below. Engine 10 will, of course, provide useful output power in the form of output shaft torque. In addition, waste heat will be recovered from the cooling and exhaust systems of engine 10 and may be employed, also in the manner to be described below, so as to improve overall system efficiency.

As will be apparent from the description of FIGS. 3—7, the present invention has several operational modes. These operational modes and the novel manner of control permits the requirements for compressor delivery to define the speed of compressor 14. It is particularly to be noted that, while generator 16 is not an essential component of the present invention, when the generator is incorporated the need for the delivery of electrical power at a constant frequency to the auxiliary devices controls the engine fuel rate simultaneously with the control of compressor speed. In order to achieve fuel control in systems incorporating generator 16, a frequency comparator unit 22 of conventional design is connected across the output terminals of the generator. In the manner known in the art, comparator 22 will provide an output signal having a polarity commensurate with the direction of and having a magnitude commensurate with the degree of generator output frequency error. This error signal is fed back to fuel control 20 via conductor 24 in such a manner that, should the speed of generator 16 exceed or fall below the proper level, the solenoid operated valve in fuel control 19 will be operated in the proper direction to regulate the supply of fuel to and thus the speed of engine 10.

As noted above, the present comfort control system contemplates a wide range of speeds of compressor 14. Obviously, at low speeds refrigerant compressor 14 would not be sufficiently lubricated from a self-contained lubrication pump as has customarily been accomplished in the prior art. Accordingly, one of the accessory electrical devices powered by the output of generator 16 will be an externally driven pump 26 for compressor 14; pump 26 supplying lubricating oil to the compressor from an oil reservoir 28 via conduit 30, the oil being returned to the reservoir via conduit 32.

In the preferred embodiment of the present invention shown in FIG. 1, the output shaft power furnished by gas engine 10 is split between the compressor 14 and generator 16 by means of a planetary differential gear arrangement 12. A preferred embodiment of this gear arrangement will be described below in the course of the description of FIG. 2. Planetary differential 12 splits the power delivered by engine 10 between the compressor 14 and generator 16 with a fixed torque ratio between the drive shafts 54 and 36, ultimately connected to the compressor and generator respectively. To achieve control, a low-power hydrostatic circuit couples the compressor drive shaft to the generator drive shaft and modifies the net torque to the individual requirements of generator 16 and compressor 14. This hydrostatic circuit comprises a variable displacement pump 38 which is driven from the shaft 36 of generator 16. Pump 38 supplies high pressure oil from a reservoir 40 to a fixed displacement hydraulic motor 42. Hydraulic motor 42 is coupled to the compressor shaft in the manner to be described below via an idler gear 44.

Referring now to FIG. 2, it can be seen that the connection between gas engine 10 and planetary differential 12 is by means of an engine fly wheel 46 which is suitably modified to also serve as a carrier for a 20-tooth planet gear 48. Planet gear 48 is rotatably mounted on shaft 50 extending from fly wheel 46 and is in driving engagement with a 20-tooth sun gear 52 which is secured to generator drive shaft 36. A 60-tooth ring gear 54 is rotatably mounted on shaft 36 by suitable bearings, not shown, and the interior teeth of ring gear 54 are in driving engagement with planet gear 48. The exterior teeth or ring gear 54 are in driving engagement with idler gear 44 which is mounted on the output shaft of hydraulic motor 42. Idler gear 44 is also in driving engagement with a gear 56 mounted on shaft 34 of compressor 14.

As fly wheel 46 rotates, planet gear 48 is moved with respect to the interior teeth of ring gear 54 and in turn drives sun gear 52 whereby generator shaft 36 is turned and the generator is driven. Variable displacement hydraulic pump 38 is drivingly connected to generator shaft 36 via a direct gear train indicated generally at 58, and variable displacement hydraulic pump 38 supplies high pressure oil to fixed displacement hydraulic motor 42. The displacement of pump 38 is regulated by an independent control which is responsive to evaporator conditions. This independent control comprises a temperature sensor 60, which is operatively associated with evaporator 18, and a servomotor 62. The output shaft 64 of servomotor 62 is connected to pump 38 and produces changes in the displacement of pump 38, such displacement changes resulting in changes in the power transmitted to motor 42. Accordingly, the amount of power delivered to compressor shaft 34 is varied in response to the need for "cooling" which is monitored at the evaporator by sensor 60. A need for additional cooling, as indicated by an increase in the temperature monitored at evaporator 18, will be sensed by sensor 60 which generates a control signal for servomotor 62 of proper polarity to cause servomotor 62 to increase the displacement of variable hydraulic pump 38. Conversely, excessive cooling will be indicated by decreased temperatures at the evaporator and, through the action of sensor 60 and servomotor 62, a decrease in the displacement of hydraulic pump 38 will be produced.

To summarize the above-described operational control, the input power delivered by engine 10 fly wheel 46 is split between generator shaft 36 and compressor shaft 34 with the means determining the split of power being the hydrostatic control system comprised of variable displacement hydraulic pump 38 and fixed displacement hydraulic motor 42. At maximum displacement of pump 38, the greatest amount of power is transmitted to motor 44 and thence to compressor shaft 34. As the displacement of pump 38 is decreased, in response to a reduced cooling load, there will be a change in the power split between shafts 36 and 34 with the power delivered to and thus the speed of shaft 34 being proportionally decreased. Conversely, an increase in pump displacement will result in a change in the power split between shafts 36 and 34 so as to cause an increase in the power delivered to shaft 34. As noted above, generator frequency is independently controlled by monitoring generator output frequency and adjusting the engine fuel rate and thus engine speed independently of the power split between shafts 36 and 34.

Referring now to FIGS. 3—7, the heating capabilities of the present invention, in the form of energy flow diagrams, are presented. Thus, considering first FIG. 3, when no cooling is required the operation of the gas engine 10 to supply ventilation and circulation power to the fans which comprise a load on generator 16 results in the generation of significant waste heat which can be recovered and delivered via conduit 65 (FIG. 1) to heat the structure in which the system is installed. Conduit 65 carries the transport material which receives the waste heat from the engine coolant heat exchanger (not shown) and from the engine exhaust recovery heat exchanger (also not shown). The transport material will, in the usual instance, be water as is also employed in the system chiller circuit for cooling the structure. However, the transport material could also be air in cases where cold air would be supplied to the conditioned space from a direct expansion evaporator. Under the conditions of FIG. 3, since no cooling is required, the variable displacement pump 38 will be adjusted via its control in such a manner that no power is delivered to shaft 34 of compressor 14. As represented in FIG. 3, engine 10 is providing approximately 40 percent of its rated load and about 55 percent of the energy released by the fuel burned in gas engine 10 will be transferred to the material flowing in conduit 65. The foregoing will be true regardless of whether the conditioned space needs as much heating as can be obtained from the recoverable waste heat. Accordingly, means will be provided in conduit 65 for directing the unneeded energy to an external heat exchanger, such as a cooling tower if the transport material is water, if excess heat is generated.

If controls in the heating system sense that more heat is needed than can be delivered in the mode of operation depicted in FIG. 3, a portion of the heated transport material in conduit 65 is withdrawn via adjustable flow divider 66 (FIG. 1) and is transferred via conduit 68 to conduit 70 where it mixes with the transport material returning from the cooling circuit in the conditioned space to evaporator 18. The operation of flow divider 66 may be automatically controlled by sensor 71 and associated servomechanisms which detect a need for heat, and actuate, in turn, the means 66 in conduit 65 for diverting flow from the external heat exchanger. When all flow is diverted from the external heat exchanger, diverter valve 66 having been progressively actuated to its maximum where all flow in conduit 65 passes to conduit 68, if more heat is still needed, flow is diverted by valve 74 as will be explained subsequently.

The delivery of waste heat from the gas engine to evaporator 18 results in sensor 60 reacting as if additional cooling were needed. That is, the temperature at evaporator 18 will be increased due to the delivery of waste heat from gas engine 10 thereto and this increase in temperature will, through the action of sensor 60 and servomotor 62, cause the displacement of pump 38 to be increased. The increase in the displacement of pump 38, as described above, will increase the proportionate amount of power delivered to compressor 14, and through the action of the generator speed control, will also cause an increase in the rate of delivery of fuel to the gas engine 10. This, in turn, results in additional waste heat being generated and recovered at engine 10 and delivered to evaporator 18 whereby the system "boot-straps" itself to a state where sufficient thermal energy is recoverable from condenser 17 to supply the heat demand. The energy released at the condenser 14 is transported through conduit 72, which merges with conduit 20, thus adding the heat released from the condenser to the remaining portion of the waste heat recovered from the engine, the waste heat being increased as above described because of the increased fuel supply supplied to engine 10 to supply power to both generator and compressor.

FIG. 4 is an energy flow diagram depicting the operational mode where all of the recoverable waste heat generated by engine 10 is applied to evaporator 18 as a "false" cooling load. That is, FIG. 4 depicts the condition where flow divider 66 is adjusted such that all of the transport material flowing in conduit 65 is diverted to evaporator 18. It is to be noted that from the standpoint of heating capability, FIG. 4 depicts the limiting condition of prior art schemes such as that described in U.S. Pat. No. 3,236,293 to Carleton. It is especially to be observed that the "false" cooling load as shown in FIG. 4 is not as large as the design cooling capability of evaporator 18 unless the gas engine 10 is unusually inefficient. This means that the engine 10, under the condition of FIG. 4, will not be loaded to its fullest capacity unless additional cooling load is imposed upon the system as, for example, by a zone within the conditioned space which needs cooling while the rest of the conditioned space needs heat. Such "cool" zones are exemplified by computer locations and cafeterias and the operation of the present invention when such an additional cooling load is present is shown in FIG. 5. It is to be especially noted that the operational mode depicted in FIG. 5 was not anticipated as advantageous in the prior art. By comparison of FIGS. 4

and 5 it may be seen that the net heat delivery for the FIG. 5 operational mode is greater than FIG. 4.

One of the significant novel aspects of the present invention is its ability to operate so that the heating capability indicated in FIG. 4 is not the limit when no real cooling load is available as shown in FIG. 5. FIG. 6 shows that energy released by condenser 17 may be fed back to evaporator 18 to increase the apparent cooling load on the evaporator thereby increasing, in the manner above-described, the displacement of pump 38, and thus also increasing to a maximum the speed of compressor 14 and the fuel flow to gas engine 10. Through a comparison of FIGS. 4 and 6, it may be seen that the additional artificial cooling load provided by the energy fed back from condenser 17 provides a net increase in heat delivery of about 30 percent. The novel feedback loop concept whereby energy is fed as artificial cooling load directly from condenser 17 to evaporator 18 may be implemented by employing a further flow divider 74 in conduit 72 whereby a portion of the heat released by condenser 17 may be delivered via conduit 76 to conduit 70. Alternatively, in the interest of configurational simplicity, flow dividers 66 and 74 and their respective associated conduits 68 and 76 may be eliminated and withdrawal of energy for feedback to evaporator 18 may take place at section Y–Y of conduit 65. The latter arrangement, that is, the employment of a flow divider and suitable means for the transport of thermal energy from section Y–Y of conduit 65 to the input to evaporator 18 (conduit 70) inherently permits all degrees of false loading as depicted in FIGS. 3, 4 and 6. Control of the flow divider or dividers for the operational mode of FIG. 6 is achieved as described above with relation to FIG. 4.

FIG. 6 also represents the unique capability of the present invention for bringing engine 10 up to full load on heating when no internal cooling load is available. Typically, comfort control systems in accordance with the present invention will have a heating capacity which is 96 percent of the installed cooling capacity. It is especially to be noted that this is a significantly greater heating capacity than state of the art gas engine driven, reverse cycle, air source heat pumps and is roughly twice the heating capacity of comparable electrical air source heat pumps. As previously noted, if all of the recoverable waste heat is fed to the evaporator as a false cooling load, the maximum loading of engine 10 is as depicted in FIG. 4. Thus, again through a comparison of FIGS. 4 and 6 which are approximately to scale, it may be seen that maximum heating capacity of the engine may be achieved without an internal cooling load by means of the feedback of energy from the condenser to evaporator.

FIG. 7 schematically shows the operation at the same engine power level as depicted in FIG. 6 in the situation where cooling and heating are desired simultaneously and the net heat delivery is increased by the amount of heat not needed in the feedback loop shown in phantom on FIG. 7. If less heating were required, less of the recoverable waste heat would be used to create the regenerative cooling load, thereby reducing the required compressor power, the engine power and the fuel input rate. If the cooling requirement, unaugmented by artificial load, made more heat available than was needed by other parts of the building, the surplus heat would obviously have to be rejected. In the present invention, internal cooling requirements increase the heating capability beyond the full engine load value of 96 percent on a 1:1 basis. Thus, once the engine is fully loaded, each unit of real cooling load releases a unit of recoverable waste heat to the heating system.

It is to be observed that the feedback loop as illustrated in FIGS. 6 and 7, wherein transport material is fed back to the evaporator from the condenser, preferably after being withdrawn from conduit 65 at section Y–Y in FIG. 1, is also applicable to systems that do not incorporate a generator for the powering of comfort condition electrical auxiliaries. Where electrical generation is not incorporated in the system, as depicted in FIG. 8, the present invention provides a dramatic simplification in system controls by maintaining constant evaporator control conditions for both heating and cooling. Similarly, from the standpoint of comparison with the closest prior art equipment from an operational viewpoint, the present invention is considerably less complex and yet provides nearly as much heating as an engine driven, reverse cycle heat pump could supply at design conditions. The present invention, moreover, can deliver up to its maximum heating capability regardless of the outside temperature thus eliminating one of the glaring deficiencies in prior art reverse cycle heat pumps. It will be obvious to those skilled in the art that, with moderate outside temperatures, the classical reverse cycle heat pump will heat with a relatively smaller fuel consumption than the novel system of the present invention but the cost and complexities of reverse cycle operations have been shown in practice to far outweigh any such minor operating cost advantages. With the generator **16** eliminated, the speed of engine **10** is directly controlled by the need for refrigerant flow. That is, with generator **16** omitted, engine **10** and compressor **14** are rigidly connected and control of engine **10** is achieved conventionally with the compressor being sequentially unloaded and the engine (compressor) speed being reduced as less refrigerant flow is needed, and vice versa.

It is also to be noted that the present invention exhibits excellent modulation characteristics in that the compressor speed is infinitely variable over a wide operational range and, therefore, the heating rate is also infinitely variable over a wide range without the need for unloading valves in the compressor when electrical generation is employed. It is also to be noted that, with or without generation, in situations where more heating capacity than cooling capacity is required, the present invention may be augmented with an auxiliary boiler. In such cases, controls can be arranged to fire the boiler when full compressor speed is reached and, once the boiler heat is attained, the compressor speed would drop for equilibrium. If more heating were required, the compressor speed would be increased from the new equilibrium value accordingly; and if less heating were required, the compressor speed would decrease accordingly until, at a minimum preset compressor speed, the boiler would be turned off. As a result, the boiler would be fired at full rate for a long time as the engine system provides the load modulation over a wide net range and once turned off, the boiler would stay off for a long period. The economic advantages of such an operational mode are significant and will be obvious to those skilled in the art since boilers have long been known to be quite inefficient during the rapid on-off cycling associated with normal operation at less than full demand.

The improvements precipitated by the present invention and described in particular in FIGS. **6** and **7** will be immediately obvious to those skilled in the art through reference to FIG. **9** which comprises a pressure enthalpy diagram. On FIG. **9**, the cycle A–B′–D–E–A is a typical air-conditioning cycle with compression occurring between A and B′, cooling and condensation of the refrigerant occurring from B′ to D, throttling through the expansion valve from B to E, and evaporation of the refrigerant occurring from E to A. In FIG. **9**, the enthalpy values are in Btu per pound of refrigerant and the inlet density to the compressor and the compressor speed (and also the volumetric efficiency of the compressor) define the actual mass flow in the system. As is traditional for the charting of refrigerant properties, a saturation line separates subcooled liquid on the extreme left from the zone of mixed vapor liquid while a second saturation line separates the vapor liquid phase from superheated vapor at the extreme right. The ideal constant entropy compression process from A to B does not account for inefficiencies in the compressor which are all normally returned to the refrigerant and thus the actual compression process is shown going from A to B′. In the interests of simplicity, the entrance to the compressor at A is shown at the saturated (vapor) line while, in actual practice, the refrigerant delivered to the compressor is slightly superheated.

When the same condenser and evaporator pressure conditions are to be used in both the heating and air-conditioning modes, the importance of the feedback loop of the present invention may be clearly seen from FIG. 9. To develop the energy required for compression, as represented by the enthalpy difference A–B′, fuel energy represented by NRG–1 must be released in the engine with that portion of this energy represented by the enthalpy change F–A recoverable as waste heat from the engine. If throttling could be controlled from point C to point F, both within the "vapor dome," the cycle A–B′–C–F–A would deliver the desired heating effect. However, the inlet to the throttling process is designed to be slightly subcooled liquid, as at point D, for the air-conditioning cycle and the cycle control would have to be modified to pass the proper proportion of vapor during heating in order to accomplish the equivalent of throttling expansion C–F. Rather than accomplish the awkward control process required for expansion C–F, the present invention accomplishes the normal complete condensation process from B′ to D and uses the excess energy released from C to D to provide the needed extra evaporation energy from E to F to balance the energy flows.

The case where electrical generation is included is represented by the additional waste heat from F to F′ that can be applied to the evaporator, the similarly increased heating energy being indicated from B′ to C; and the reduced feedback energy C′–D=E–F′.

FIG. 9 also shows the case where feedback is not employed and a conventionally controlled expansion valve is utilized, this condition being depicted by the cycle *a*–*b*–D–*e*–*a*. In this case, the compressor inlet pressure is markedly reduced so that the compressor work is large enough relative to the evaporation requirement to enable the recovery of enough waste heat from the engine. The actual engine power, however, is defined by the product:

(NRG–2) ×(compressor speed) ×(density at *a*) ×(volumetric efficiency at *a*)

and is a much smaller power level than the product:

(NRG–1) ×(compressor speed) ×(density at A) × (volumetric efficiency at A)

primarily because: 3ps

$$\frac{(\text{Density at A})}{(\text{Density at a})} > \frac{(\text{NRG–2})}{(\text{NRG–1})}$$

but also because:

(volume efficiency at a) <(volume efficiency at A)

because of the increased pressure ratio. It thus may be seen that the power level represented by the normal air-conditioning compression from A–B′ is very near the maximum with a given condensing temperature and compressor speed. As a result, the best way to fully load the engine is to maintain air-conditioning evaporator and condenser conditions during heating and this is achieved by the present invention in contradistinction to prior art engine-driven, heat pump systems.

While a preferred embodiment has been shown and described, various modifications and substitutions may be made thereto without departing from the spirit and scope of the present invention. Accordingly, it is to be understood that the present invention has been described by way of illustration and not limitation.

I claim:

1. A comfort conditioning system for a building structure comprising:

refrigeration means including a condenser, a compressor and an evaporator;

a driving engine;

means coupling the mechanical output power of said engine to the compressor of said refrigeration means;

means for selectively delivering a portion of the thermal energy rejected from the condenser of said refrigeration means directly to the evaporator of said refrigeration means as an artificial cooling load to increase the thermal energy rejected by the condenser as required for heating purposes; and means for delivering the remainder of the recoverable thermal energy rejected from the condenser of said refrigeration means to the building structure for heating purposes.

2. The system of claim 1 wherein said driving engine comprises an internal combustion engine.

3. The system of claim 2 further comprising:
means for delivering a combustible fuel to said engine;
means for recovering waste heat produced in the course of combustion of fuel in said engine; and
means connected to said waste heat recovering means for delivering engine waste heat as an artificial cooling load to said refrigeration means evaporator.

4. The system of claim 3 further comprising means connected to said waste heat recovering means for delivering the recovered engine waste heat which is not employed as an artificial cooling load to the building structure for heating purposes.

5. A comfort conditioning system for a building structure comprising:
refrigeration means including a condenser, a compressor and an evaporator;
an internal combustion engine;
means for delivering a combustible fuel to said engine;
means coupling the mechanical output power of said engine to the compressor of said refrigeration means;
means connected to said engine for delivering waste heat produced in the course of combustion of fuel to the building structure for heating purposes;
means connected to said refrigeration means condenser for delivering thermal energy rejected therefrom to the building structure, the rejected thermal energy being added to the engine waste heat; and
means for selectively delivering a portion of the combined engine waste heat and condenser rejected thermal energy directly to the refrigeration means evaporator as an artificial cooling load to increase the thermal energy rejected by the condenser as required for heating purposes.

6. The system of claim 5 wherein said system further comprises means for generating electrical power, said generating means being driven by said engine.

7. The system of claim 6 wherein said power coupling means comprises means for splitting the output shaft power of said engine between said generator and said refrigeration means compressor.

8. The system of claim 7 further comprising:
means for sensing a condition at the refrigeration means evaporator and for generating a signal commensurate therewith;
means connected to said power splitting means for varying the amount of mechanical power delivered respectively to said generator and compressor; and
means responsive to said signal commensurate with an evaporator condition for causing adjustment of said power split varying means.

9. The apparatus of claim 8 wherein said power splitting means comprises planetary differential gear means.

10. The system of claim 4 wherein said system further comprises means for generating electrical power, said generating means being driven by said engine.

11. The system of claim 10 wherein said power coupling means comprises means for splitting the output shaft power of said engine between said generator and said refrigeration means compressor.

12. The system of claim 11 further comprising:
means for sensing a condition at the refrigeration means evaporator and for generating a signal commensurate therewith;
means connected to said power splitting means for varying the amount of mechanical power delivered respectively to said generator and compressor; and
means responsive to said signal commensurate with an evaporator condition for causing adjustment of said power split varying means.

13. The apparatus of claim 12 wherein said power splitting means comprises planetary differential gear means.

14. A method of comfort conditioning a building structure comprising:
operating a refrigerating system in the cooling mode of comfort conditioning;
continuing to operate the refrigerating system without reversal of the refrigerant cycle during the heating mode of comfort conditioning;
delivering thermal energy rejected from the condenser of the refrigerating system to the building structure during the heating mode; and
delivering a portion of the thermal energy rejected from the condenser of the refrigerating system to the evaporator of the refrigerating system as an artificial cooling load during the heating mode.

15. The method of claim 14 further comprising:
driving the refrigerating system compressor with an internal combustion engine;
recovering waste heat produced during operation of the engine; and
delivering recovered engine waste heat to the building structure during the heating mode.

16. The process of claim 15 wherein the condenser rejected thermal energy is added to the recoverable engine waste heat in the heating mode and wherein the step of delivering thermal energy as an artificial cooling load comprises diverting a portion of the combined engine waste and condenser rejected thermal energy from the building structure to the refrigerant system evaporator.

17. The method of claim 15 further comprising driving an electrical generator from the engine which drives the refrigerant system compressor.

18. The method of claim 17 wherein the step of driving the generator comprises splitting the engine output shaft power between the compressor and generator.

19. The method of claim 18 further comprising:
sensing a condition at the refrigerant system evaporator; and
controlling the amount of power delivered respectively to the compressor and generator in accordance with the sensed condition.

20. The method of claim 16 further comprising driving an electrical generator from the engine which drives the refrigerant system compressor.

21. The method of claim 20 wherein the step of driving the generator comprises splitting the engine output shaft power between the compressor and generator.

22. The method of claim 21 further comprising:
sensing a condition at the refrigerant system evaporator; and
controlling the amount of power delivered respectively to the compressor and generator in accordance with the sensed condition.

23. The method of claim 19 further comprising:
sensing the electrical generator output frequency; and
adjusting the flow of fuel to the engine to regulate engine speed to thereby control generator frequency.

24. The method of claim 28 further comprising:
sensing the electrical generator output frequency; and
adjusting the flow of fuel to the engine to regulate engine speed to thereby control generator frequency.